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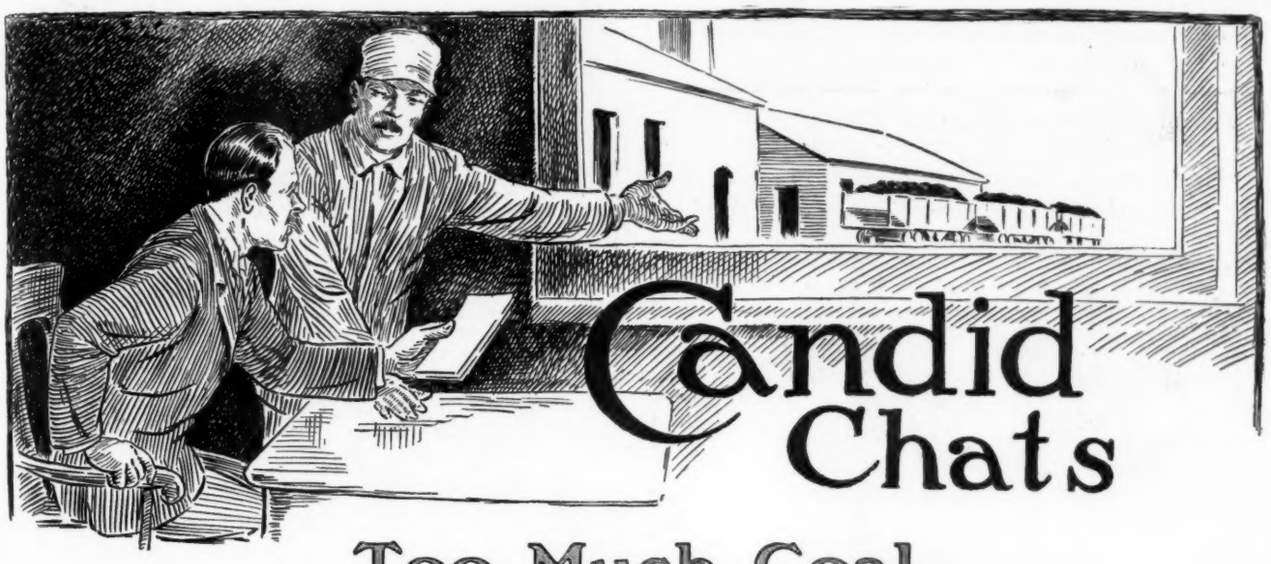


Candid Chats

Too Much Coal

Here is an amusing yet pathetic story which you'll appreciate all the more if you know of a similar case

THE plant is the ordinary large, industrial kind where much low-pressure steam is used for manufacturing purposes. The management—the president and the usual outfit—is representative of the average.

The new chief engineer was engaged on a salary and percentage-of-saving basis. Of course, the first thing he did was to return to the boiler the many pounds of 210 deg. F. condensate allowed to merrily flow to the sump and sewer. Winter was coming on, and one need not be a college professor to know that changes in pipe sizes and arrangements and the installation of a few traps on the heating system would make the coal man feel badly.

The volume of business was fairly constant, and had been for years, so the coal was bought by contract and supplied in unvarying amounts as regularly as you get sleepy between 2 and 3 a.m. when you are on the 12-to-8 watch. The coal was stored in a large low building and had to be trimmed considerably to fill it. Soon the effect of the changes in the uses of steam and the disposition of condensate was made apparent by the full coal cars on the siding and the labor necessary to trim coal in the storage bin. The amount of coal on hand was becoming a veritable nuisance!

The chief asked the manager to have the supply stopped for a while. This was a new one on the manager; the engineer's complaint had always been the other way, and the manager did not understand. Couldn't the engineer find room for it somewhere? The quantity received was as usual, and he could prove it by comparing the monthly statements. The mills ran full time; the winter was just as cold as previous ones, if not colder. He glanced at the calendar to make sure it was Feb. 1 instead of Apr. 1, and finally walked to the window to see that string of cars. Yes, the coal was there! Ah! he had it; the coal was not being used!

The engineer suggested that they compare the weekly coal consumption reports. Yes, the manager was right; the coal was not being used as formerly. The reports showed that up to date about one hundred tons less coal had been used these last winter months than for the corresponding months of previous years.

What does this case show? Here is an actual, honest-to-goodness example of the amount of worth-while attention some concerns pay to the power plant.

And the engineer was engaged on a percentage-of-saving basis, too!

It shows that the engineer must keep duplicate reports and, above all, see to it that the management really studies them and is given every opportunity to know what their proper interpretation means in its relation to the chief engineer and the cost of manufacture.

This engineer had applied himself to the task of reducing costs and improving service, had stayed up late nights outlining necessary alterations to this end—he was making good! And he was getting no more moral credit than the man who would have used the 100 tons and saved disturbing the manager. Things would have gone this way until the books were examined to determine how much in percentage-of-saving was due the engineer.

Seattle Municipal Lighting Plant

By W. L. Kidston

SYNOPSIS—This plant will assist in carrying the winter peak loads of the Cedar Falls (Wash.) hydro-electric plant. A steam pressure of 200 lb. is carried on the three boilers, with 125-deg. superheat. The turbo-generator is of 7500-kw. capacity. The boiler furnaces are designed for burning fuel oil or coal with mechanical stokers, which can easily be put in place.

The new steam-generating station of the Seattle (Wash.) municipal system, known as the Lake Union auxiliary, has a continuous capacity of 9375 kw. and will be used by the lighting department to help the main hydro-electric station at Cedar Falls over the heavy winter peaks and to take its full capacity load in case of accident to the water-power station or transmission lines. The steam plant is near the geographic center of the city, on the east shore of Lake Union. It will be accessible from Puget Sound and Lake Washington through the Lake Washington canal, and by land it can be reached by the Lake Union belt-line railway and by the tracks of the local traction company.

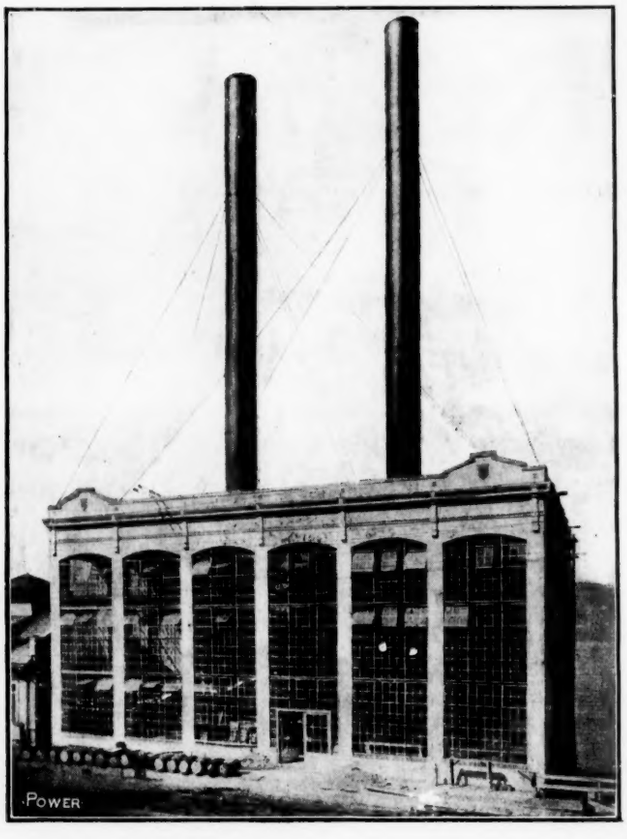


Fig. 1. New Seattle Generating Station

The building (Fig. 1), built of reinforced concrete, is 90x100 ft. and 57 ft. from the basement floor to the cornice and was begun on Apr. 25 of last year. It was designed and built by the Department of Buildings of the city of Seattle. The foundation is on piles. The basement floor is 18 ft. below the level of East Lake Ave., and a concrete retaining wall containing 800 cu.yd. was built to support the street and protect the structure from possible slides from the hill beyond. Steel sash is used throughout the building, and as the space between columns is glazed, ample light is secured. A 25-ton crane with a 30-ft. span serves the turbine room.

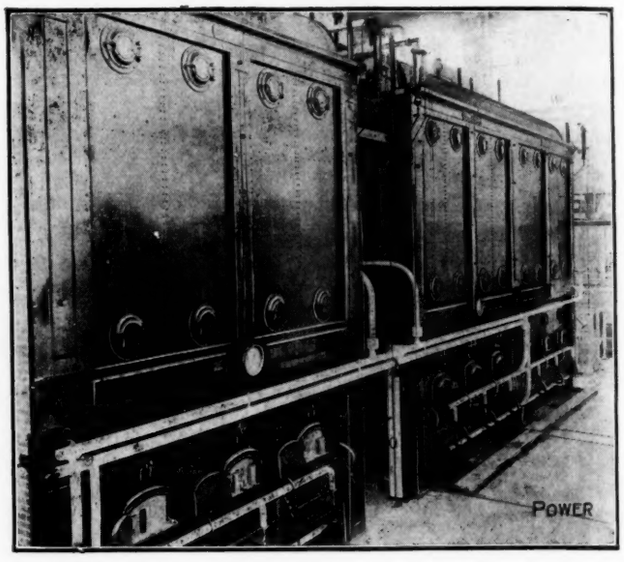


Fig. 2. Three Water-Tube Boilers with 8230-Sq.Ft. Heating Surface

The plant contains three water-tube boilers (Fig. 2), each having 8230 sq.ft. of heating surface, which will supply steam at 200 lb. pressure and 125 deg. superheat to the 7500-kw. turbo-generator set (Fig. 3). The boilers are on the street floor at the west side of the building, with the firing aisle next to the lake. They are equipped for burning oil, but the settings are arranged for stokers and ash hoppers, so that a change may be made to coal burning at any time by inserting the stokers, the tracks for which are in place. The basement under the boilers is planned to accommodate the ash-handling cars. Provision is made for a fourth boiler. The boilers are guar-

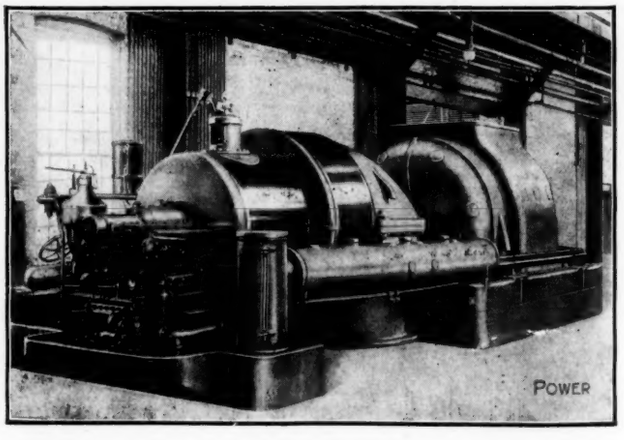


Fig. 3. The 7500-Kw. Turbo-Generator Unit

anteed for 77 per cent. efficiency at full load and to operate satisfactorily at 180 per cent. continuous overload. Two steel stacks, 90 in. diameter, designed for coal burning, extend 170 ft. above the boiler-room floor, each to care for two boilers.

Two steel oil-storage tanks, 11 ft. and 20 ft. long, of 15,000 gal. capacity each, are buried just outside the boiler room on the south side of the building. Both tanks have domes 4 ft. in diameter and 4 ft. high, with screw covers and steel ladders to permit of inspection. This capacity per tank is the largest allowed by the city ordinance. As the tanks are adjacent to the building, it was necessary to bury them, and a concrete wall was built, separating and inclosing them. The space around the tanks was then filled with earth to a depth of 4 ft. above the top of the shells. Six-inch connections to the domes are used for filling the tanks and connections of the same size on the under side join with an 8-in. suction header, to which the two motor-driven storage pumps, each with a capacity of 16,000 gal. per hour, are connected. This header runs out to the lake for use in unloading oil-tank cars or oil scows.

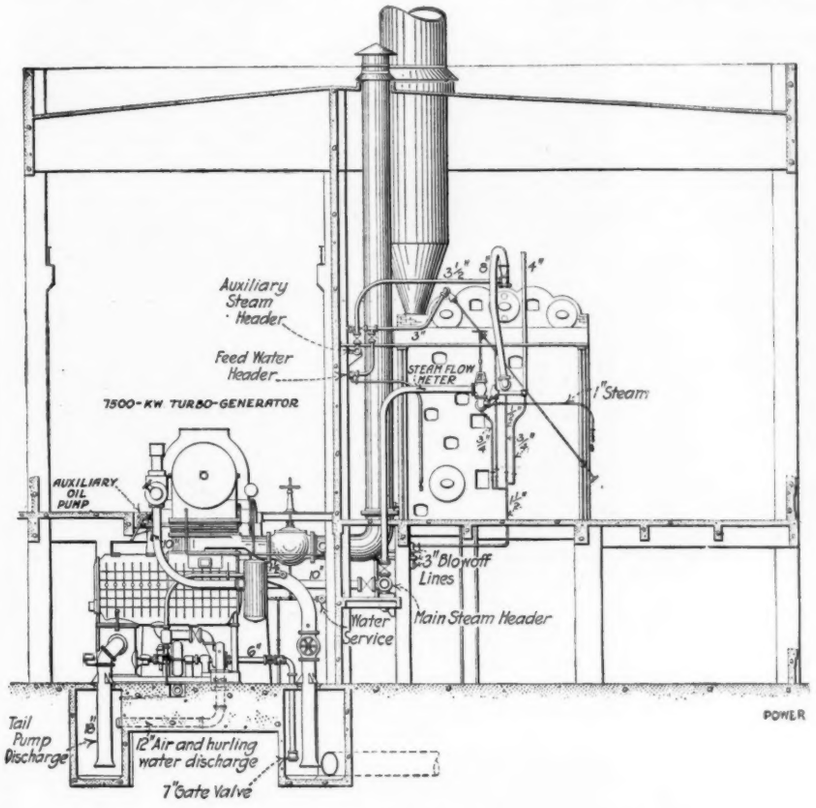


FIG. 4. ELEVATION OF THE BOILER ROOM AND BASEMENT

One 7200-gal. service tank, 7 ft. in diameter by 24 ft. long, is placed above the storage tanks, and the suction pipes from the two burner pumps run through the dome and down to the bottom of the tank, terminating in a foot valve. The tank is separated into halves by a partition, thus forming two tanks, both of which have a 30-in. dome with a screw cover. A steel ladder runs to the bottom of the tank. The 6-in. connections in the domes are used for filling the tanks and the 4½-in. suction pipes for emptying them. A 2½-in. overflow pipe runs from each oil heater back to the service tank. A 6-in. pipe from the filling pipes to the street makes it possible to fill any of the three tanks from oil trucks. Pumps to supply the oil burners are on the boiler-room floor, as are the oil heaters.

The boilers connect to a 12-in. header (Fig. 4), from which steam is taken to the turbine (Fig. 7). The generating unit is a horizontal turbine connected by a flexible coupling to a 2500-volt, two-phase alternator, at 1800 r.p.m., and rated at 7500 kw. at 80 per cent. power factor with 50-deg. Centigrade rise above the room temperature, and at 9375 kw. at 80 per cent. power factor with 65-deg.

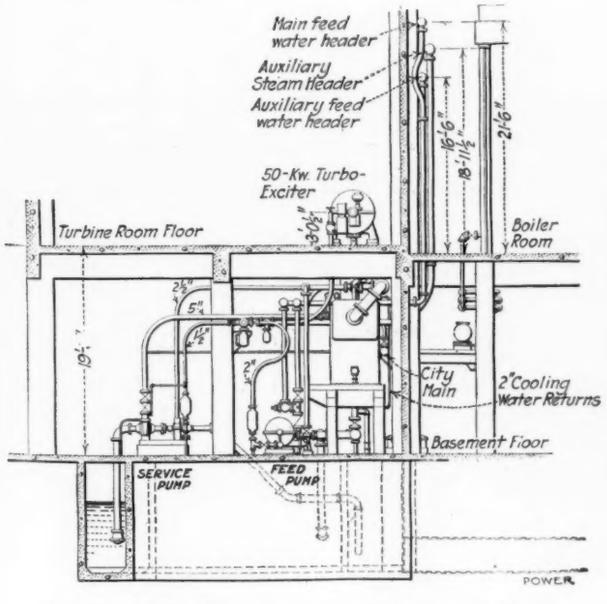


FIG. 5. END ELEVATION OF AUXILIARY PIPING

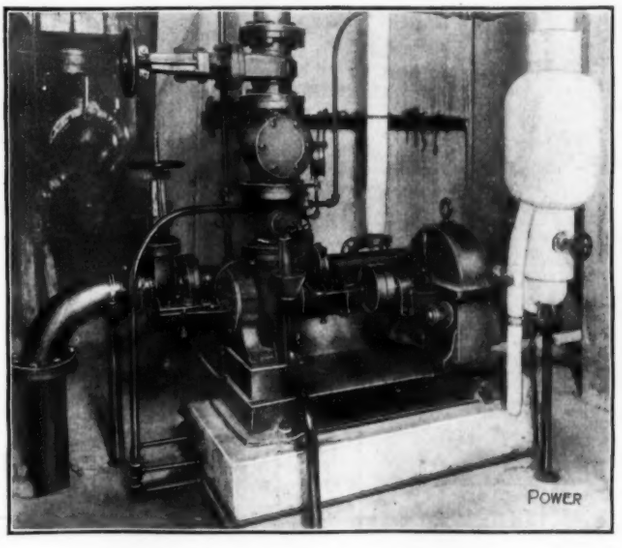


FIG. 6. TURBINE-DRIVEN HOUSE PUMP

rise. The temperature in the generator coils is measured by a resistance thermometer inserted between them, and is registered on the switchboard. The unit is guaranteed to produce one kilowatt-hour from 12.95 lb. of steam at 190 lb. pressure and 125 deg. superheat when operating at full load.

The condenser, of the rectangular-jet type, and placed under the turbine (Figs. 4 and 5), will maintain 28½ in. of vacuum when condensing 97,500 lb. of steam per hour.

The centrifugal circulating pump is mounted on the shaft with the rotary air pump and both are driven by a small impulse turbine. The boiler-feed pumps and the service pump (Figs. 6 and 9) are also of the centrifugal type with four stages, and are turbine driven. Steam for these auxiliaries is taken from a 6-in. saturated-steam header, and the exhaust is used in the 2500-hp. metering feed-water heater. The exciter, rated at 50 kw., 125

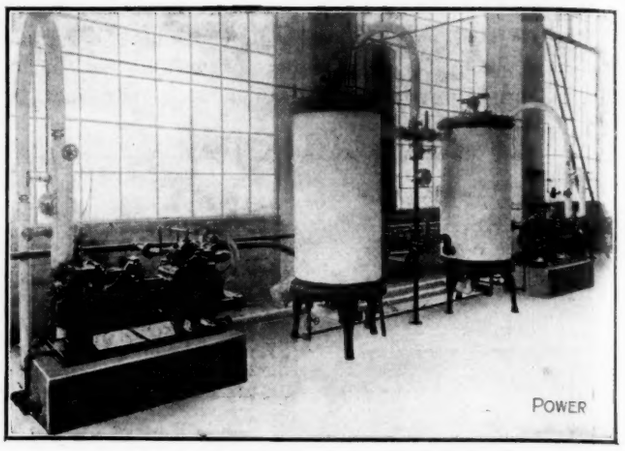
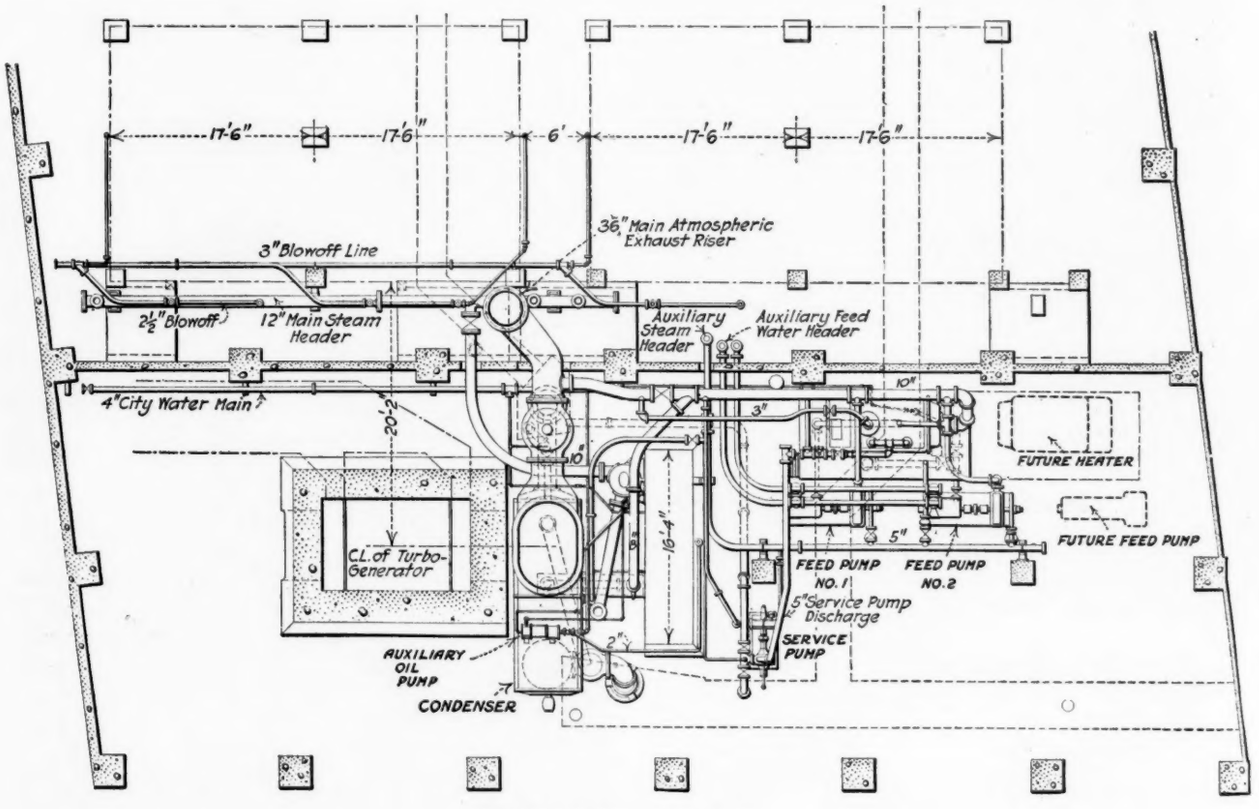
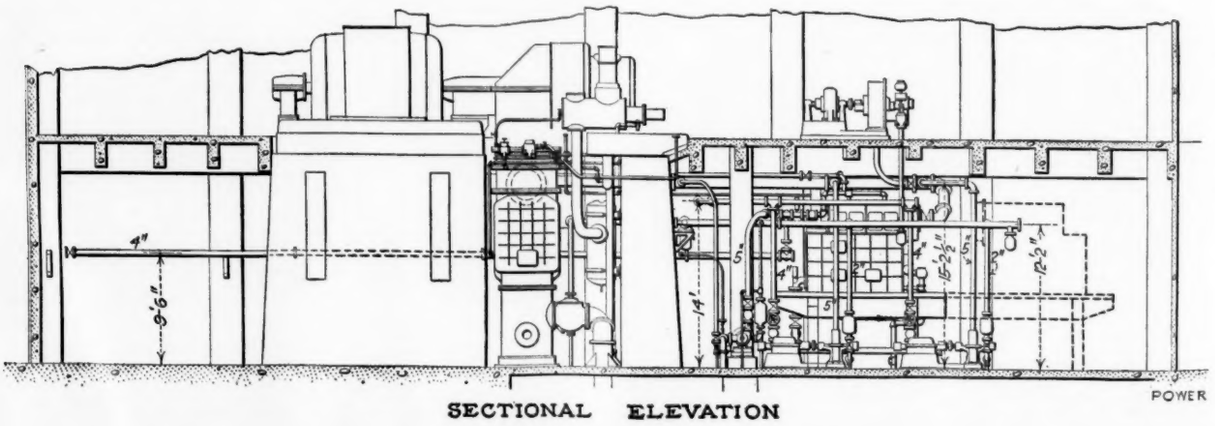


FIG. 7. OIL HEATERS AND PUMPS



BASEMENT PLAN



SECTIONAL ELEVATION

FIG. 8. PLAN AND ELEVATION OF TURBINE-ROOM BASEMENT

volts, is also turbine driven. Fig. 8 shows the piping arrangement.

As the steam-plant site is on the shore of Lake Union, a fresh-water lake, there is an abundant supply of cooling water for the condenser. The water supply is brought to the plant from an intake 120 ft. out in the lake, through a 30-in. cast-iron pipe to a concrete screen box at the west side of the building, and from there through a second run of pipe to a cold well at the end of the condenser. An 18-in. pipe supplies the condenser with cooling water, which is drawn into the condenser by vacuum. In starting, a jet of water from the city mains is used to condense

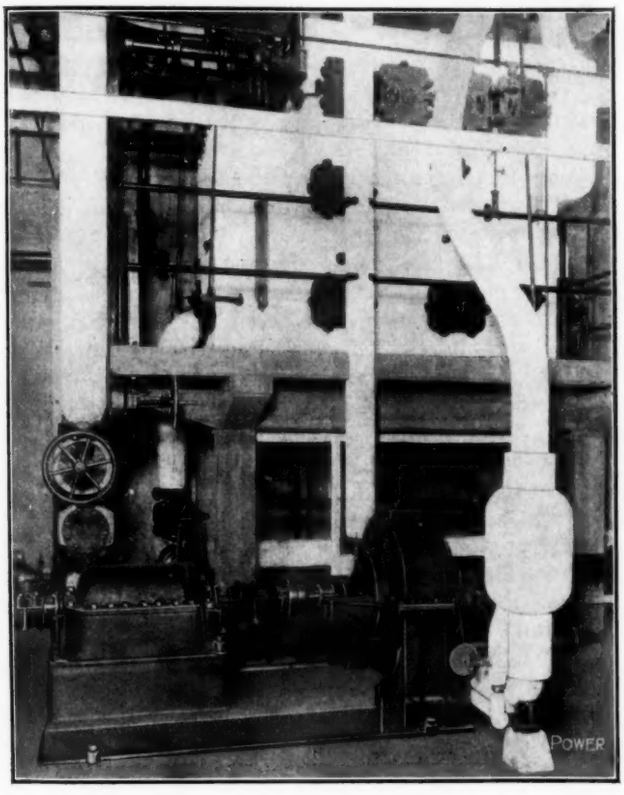


FIG. 9. ONE OF THE TWO TURBINE-DRIVEN CENTRIFUGAL BOILER-FEED PUMPS

the steam and create a vacuum in the condenser shell. The 18-in. discharge pipe from the condenser connects with the hotwell, which is a concrete tunnel 4 ft. wide by 10 ft. deep. In case of a future installation, this tunnel will connect the two units. A 30-in. cast-iron pipe serves as an outlet for the hotwell and discharges the hot water into the lake at the back wall of the building.

The turbine set is on the street floor, on the east side, next to East Lake Ave., and the switchboard is in the same room. Current is stepped up to 15,000 volts, two-phase, for distribution over the city, and the steam station will be connected to the main distributing station at Seventh Ave. and Yesler Way by a direct 15,000-volt tie line. The step-up transformers, of the same capacity as the turbine, are in the basement under the turbine room, where the oil switches and 2500-volt and 15,000-volt wiring are placed. Provision is also made to use the steam station for distributing, for which purpose the oil switches, feeder regulators and street-lighting transformers will go in the basement.

The steam plant is on the same lot with a 1500-kw. water-power auxiliary, which uses the overflow from the Volunteer Park reservoir of the city water system, situated on the hill 415 ft. higher than the lake. Both plants will be operated from the same switchboard and will work together to safeguard the service of the system.

The main generating station at Cedar Falls is being extended and improved by the erection of a $1,400,000 dam, which will permit of a development of 40,000 kw. J. D. Ross, superintendent of lighting, is in charge of the Seattle municipal plant.

First-Aid Jar for Power Plants

Accidents frequently happen in and about a power plant, and in many cases the injured could be attended by the laymen if first-aid materials were at hand. A first-aid jar has been prepared to meet such requirements by the Conference Board of Safety and Sanitation, of which Magnus W. Alexander, General Electric Co., West Lynn, Mass., is secretary.

The jar is structurally strong and a special annealing treatment makes the glass still stronger. It is made with smooth surfaces and with straight walls on the inside to promote cleanliness and facilitate the removal of first-aid materials. A convenient carrying handle is molded to the glass cover, held by suitable spring clips which are a part of a metal cage holding the jar; this cage affords added protection against breakage. A rubber gasket between the jar and the cover makes the outfit dustproof.

The jar is made only high enough to accommodate the bottles of medicaments stored in it so that the stoppers

FIRST-AID JAR

cannot come out when the cover rests on the jar. Medicine bottles, bandages, absorbent cotton, burn ointment in collapsible tubes and a wire-gauze splint are arranged along the wall of the jar, so that they are plainly visible from the outside and can be quickly located. A specially constructed metal dish placed inside of the jar keeps the materials in their proper places; it is also used as a receptacle for tourniquet, medicine glass, gauze bandages, medicine droppers, spoon, scissors, etc.

The jar is about 9½ in. in diameter, 6 in. high, and

complete with contents weighs only slightly more than 12 lb. It, however, includes every material which a conference of physicians with extensive experience in the treatment of injuries agreed upon as necessary for effective first-aid treatment by laymen.

Suitable first-aid instructions are printed on the inside of the cover, while on the outside appears the standard list of materials which should always be kept in the jar and brief directions for the use and care of the outfit.

This jar has been approved by the board, which is composed of representatives of the National Founders' Association, 29 South La Salle St., Chicago; the National Association of Manufacturers, 30 Church St., New York City; the National Metal Trades Association, Peoples Gas Bldg., Chicago, and the National Electric Light Association, 29 West 39th St., New York City. The outfit is sold at practically cost price, as there is no intention to make a profit on any of the articles standardized by these associations, and can be secured by writing the secretary of any of the associations mentioned.

Oil-Engine Tendencies

By A. E. Ward

SYNOPSIS—The writer reviews the oil-fuel situation and points out the legitimate fields for the different types of engines. He warns against the defects in the low-compression, pump-injection type when attempting to use heavy oils, which should be used only in the high-compression engine, and champions the vaporizing type.

The manufacture of internal-combustion engines is being influenced to a certain extent by the fuel situation, and the consequent demand of the public for an engine which will handle heavy grades of liquid fuels. Whether this demand is to continue will be influenced by the availability of the heavy liquid fuels, their prices as compared with lighter fuels, and the success of the engines which claim to use them. Therefore, it would be well for the buying public to consider carefully and without prejudice all phases of this subject before it continues to bring about what may prove an undesirable tendency in the application of the internal-combustion engine for general power purposes.

Oil-Fuel Situation

During the past few years there have been marked changes in the oil-fuel situation, and it is practically impossible to prophesy what the future will develop. The heavy drain on the gasoline supply for the automobile trade has been the apparent cause for its rise in price. However, this has now been reduced to practically the figures which prevailed several years ago, but it is still too high to be considered for use in the larger or intermediate sizes of internal-combustion engines.

An important feature is the percentage of fuel of different grades available from crude oil. Refiners have been able to secure in recent years a larger percentage of gasoline from crude oil than heretofore, but the quality of the gasoline has also been reduced. Crude oil from different localities varies greatly in quality, but a fair average would indicate that about 15 per cent. can be turned into gasoline or naphtha. About 45 per cent. is kerosene, and about 10 per cent. of a high-grade distillate above 39 deg.; another 10 per cent. is a low-grade distillate below 39 deg; while about 15 per cent. is turned into lubricating oil, and the remaining 5 per cent. is slop. It is evident, therefore, that a small percentage of the refined product, about 10 per cent., is of such a nature as to require a crude-oil engine. Furthermore, about 50 per cent. of the refined product can be handled in the conventional four-stroke-cycle vaporizing type.

There is a large market for the heavier grades of refined oil to be used for burning under boilers, oiling streets, etc., which often makes it difficult for the small purchaser to secure, or to continue to secure, this fuel for power purposes. Sometimes the small purchaser finds that it is necessary to buy his fuel in tank-car lots in order to gain the point of economy desired. It is evident, therefore, that the purchaser of an engine designed for handling crude oil or an equivalent fuel is necessarily subjected to the caprices of the market.

It is true that there are a number of localities, especially in the West and Southwest, where the heavy liquid fuels are being purchased at comparatively low prices. However, it is questionable whether this condition will continue, inasmuch as the supply of fuel oil to burn under boilers has frequently been taken away from manufacturers who have gone to the expense of installing boiler appliances for burning this fuel. Moreover, there is wide variation in the quality and constituents of heavy oils, depending upon the quality of the crude, the method of refining, tendency of the refineries in accordance with the demands, etc.

While gasoline is also uncertain in its price, kerosene has remained at practically the same price, has always been available in most localities, and does not vary in quality.

There are, undoubtedly, many cases where the Diesel engine is the proper type to use, considering especially its high thermal efficiency and the ease with which its fuel can be stored, transported and handled. The following objections can be raised against it, however. It is complicated in design, necessitating strict attention to the minutest details and requires a very high grade of workmanship. Moreover, correct adjustment must be maintained at all times, and skilled attendance with corresponding high cost is essential. Fuels of low quality and low price have frequently been used when the plant was first installed, and later a high-grade fuel has been substituted. Depreciation and maintenance have in many cases been excessive in addition to the first cost being high. Finally, this type of engine is extremely sensitive to irregular or improper conditions.

In spite of these objections there are many successful Diesel installations. If a purchaser takes these points into consideration, arranges to keep the engine in prime condition at all times, is assured that the proper fuel will

be available at a low price, if natural gas is not available, and his conditions do not favor producer gas, and if his power requirements are not less than 100 hp., he may be justified in purchasing a Diesel engine.

Semi-Diesel Engines

Coming now to a class of engines for powers of 15 to 100 hp., the purchasers of these sizes ordinarily do not give the engineering features due consideration. Being without the necessary experience and knowledge themselves, they are largely at the mercy of the ambitious salesman who has a tendency to exaggerate the qualifications of the article he is handling. There has been a clamor for engines of those sizes which will burn heavy fuels, such as have been successfully handled in the Diesel engine. Owing to this demand, there has been a natural response on the part of some manufacturers to produce

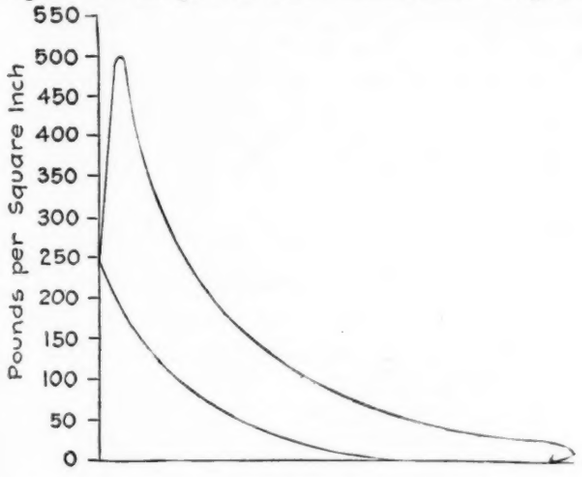


Fig. 1. Diagram from Hot-Plate, Air-Injection Engine at Rated Load

an engine which would fulfill these requirements. It is evident that the high first cost of the Diesel engine would prohibit its sale among the majority of purchasers who desire engines of this size.

There is said to be from 400,000 to 500,000 hp. in engines of the Diesel type used in Europe. There are also a number of these engines in this country, but the fuel situation in Europe is such as to necessitate the use of an engine of this type to a far greater extent than has been the case in this country. It has been repeatedly reported in this country that in engine installations of the Diesel type it has been found desirable to use a lighter grade of fuel than was originally intended. This is brought about by the fact that less trouble and less close attention are required with the lighter and higher-grade fuels. This being the case with an engine which employs extreme methods in order to burn the fuels economically and satisfactorily, how can it be at all satisfactory to handle such fuels in an engine where these methods are not employed?

In an attempt to reduce the cost of manufacture, and still satisfy the demands for an engine which will run on heavy liquid fuels, attempts have been made to depart from the Diesel principle. The first attempt was to change the compression from 500 to 300 lb. This reduction in compression means that the fuel does not burn immediately upon entering the combustion space, as the heat of compression is not sufficient, and in order to obtain the desired temperature a hot plate is projected into the cylinder.

This type retains the air system of fuel injection. A two-stage air compressor is used, which discharges at approximately 600 lb. directly through the fuel valve and against the hot plate in the cylinder. The quantity of fuel is measured in a manner similar to that of the Diesel engine. This system avoids the necessity of carrying such high injection air pressure, but there is only a partial burning of the fuel, and an explosion takes place, the initial pressure depending on various conditions, such as the timing of the fuel injection, the nature of the fuel, the temperature of the engine, the temperature of the hot plate and the compression temperature. An indicator diagram from such an engine is shown in Fig. 1.

If an engine of this type is built as heavy as it should be to withstand the high pressures, and if all other parts are properly designed and constructed, the first cost is almost as high as the Diesel.

An extreme departure from the Diesel principle is the two-stroke-cycle engine using the hot bulb, pump fuel injection, water injection, in the majority of cases crank-case compression, and light construction with a compression ranging from 70 to 150 lb., and in some cases 300 lb. In order to start this engine it is necessary first to heat the hot bulb externally, which requires ordinarily from 15 to 20 min. Sometimes difficulty with the burners necessitates a much longer time.

Two-Stroke-Cycle Principle

The main reason for the four-stroke cycle having been so universally adopted is that the scavenging of the burnt gases can be accomplished by one stroke of the main piston. This leaves the entire volume swept by the piston free for a fresh charge.

The two-stroke-cycle engine which draws the fuel into the crank case and depends upon scavenging the burnt gases by means of air is necessarily uneconomical, because it is difficult to cut off the exhaust gases at exactly the proper point and avoid all passage of the fuel through the exhaust port. However, when the fuel is injected on the compression stroke, and pure air only is drawn into the crank case, this objection is not so serious. Nevertheless, it is difficult to determine just what air currents take place inside the cylinder when it is expected that the incoming air on the one side will drive out the exhaust gases on the other, especially when both of the ports are at one end of the cylinder. Fig. 2 illustrates what probably happens in such a case.

Fuel Injection

Now, consider the fuel injection and the method of forming an explosive mixture. The conventional vaporizing type of four-stroke-cycle engine draws in a charge of air past a fuel spray nozzle at a high velocity, frequently as high as 12,000 to 15,000 ft. per min. This high velocity, together with the extremely fine spraying of the fuel, causes each particle of air to become laden with a certain amount of fuel vapor, and thus promises a very complete mixture.

On the other hand, in the engine with pump injection the combustion space is filled with a mixed charge of air and exhaust gases, with probably some irregular stratification of the exhaust gases. The fuel pump

measures out a given quantity of fuel, which is sprayed into the compressed air by a hammer blow from a cam or eccentric which operates the pump. A certain mixture of fuel with the air and burnt gases takes place, but a thorough mixture of the fuel with the air does not exist (see Fig. 3). Furthermore, this mixture will vary greatly from one impulse to the next, depending upon conditions such as stratification of the charge, temperature of the hot bulb, temperature of the engine itself, nature of the fuel used, amount of fuel measured out and injected by the pump, size of the opening in the spray nozzle, condition of the spray nozzle, time in the cycle at which the fuel is injected, etc. All conditions being uniform, the results are fairly satisfactory, but there is necessarily susceptibility to these varying conditions.

These engines are ordinarily rated at a mean effective pressure of 35 to 40 lb., or about half that of a four-stroke-cycle engine of the vaporizing type at rated load. Indicator diagrams from properly designed four-stroke-cycle engines of the vaporizing type show that the mean effective pressures are practically uniform, although in some cases a pressure of 118 to 120 lb. may be indicated. Fig. 4 is a diagram from a two-stroke-cycle vaporizing engine at rated load. Mean effective pressures as high as this are extremely rare, however, and could not be depended upon. This point is brought out to show that the mean effective pressures from the two-stroke-cycle hot-bulb engines are necessarily variable. Furthermore, it is possible to obtain extremely high initial pressure if water injection is not used properly, if the fuel is injected a little too early, or if the temperature of the hot bulb is not correct. Considering that it is possible for these high pressures to exist, the engine should be built heavy to withstand them; irregular impulses mean uncertain service. Fig 5 shows these varying mean effective pressures.

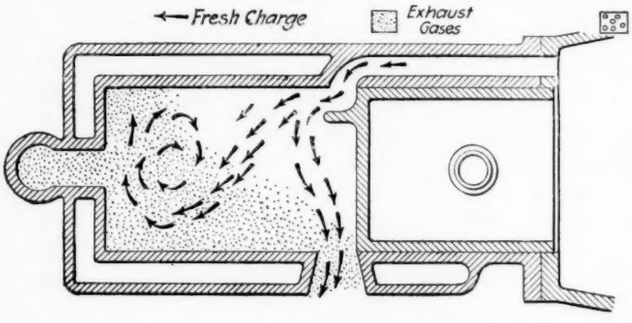


FIG. 2. RESULTS IN TWO-STROKE-CYCLE EXHAUST-SCAVENGING TYPE OF ENGINE

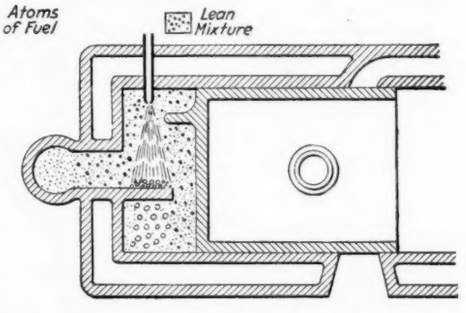


FIG. 3. SHOWING SMALL VOLUME OF AIR WITH WHICH FUEL COMES IN CONTACT IN PUMP-INJECTION TYPE

MEASURING THE FUEL

Next consider the method of measuring the fuel. Some engines govern by changing the stroke of the pump; others through bypassing a certain amount of the fuel back to the tank. If six drops per stroke is a full load supply for a 10-hp. engine, it will readily be seen how delicate the adjustment must be to govern the speed by reducing the proportion of the six drops in accordance with the load conditions.

If the vaporizing type of engine is properly designed, the correct amount of fuel will be automatically picked up as the throttled air passes over the injection nozzle. This is so automatic that the results are dependable even under adverse conditions.

TIMING OF FUEL INJECTION

Different manufacturers commence injecting the fuel at different points in the stroke varying from the beginning of the compression stroke to about 10 deg. before the end of the compression stroke. The proper timing depends upon numerous conditions, such as compression pressure, temperature of the hot bulb, shape of the hot bulb and combustion space, location of the spray nozzle, grade of fuel, diameter of pump plunger, stroke of pump, etc.

Electric ignition is practically instantaneous, and if there is a variation of say 10 deg. in timing, the results obtained are apt to be poor. With an engine running at 300 r.p.m., 10 deg. is equivalent to 0.002 sec. This is an extremely short time for a mechanical fuel pump to act and deliver a quantity of fuel against 150 lb.

It is difficult for anyone who has not worked with them to appreciate the delicacy of these mechanical devices. If the injection of fuel takes place slightly early, in comparison with the temperature of the hot bulb, the amount of water injection, the temperature of the engine and the load, etc., excessive pressure will result.

THE HOT BULB

The mission of the hot bulb is to furnish the high temperature to assist in vaporizing heavy fuels. If its temperature becomes too high, the engine will pound heavily, due to excessive initial pressure. The hot bulb will sometimes have a tendency to crack, due to distortion, because of the unequal temperatures, and also the fuel may decompose and form deposits on the hot spoon, so as to cause cracking of the bulb. Because of this the hot bulb is occasionally the source of annoyance and irregular operation.

WATER INJECTION

If the injection of the fuel could be accurately timed, so as to bring about the proper flame propagation, and if this fuel could be introduced into the cylinder in sufficiently large quantities in the very short time existing at the end of the compression stroke to prevent too early ignition, water injection would not be necessary. Engines that use air injection with the fuel do not use water. Water injection is the handiest means of avoiding excessive pounding and high initial pressure, provided the timing of the fuel injection and the temperature of the

hot bulb, etc., are not correct. It is frequently used in large quantities, and when the quantity is too great excessive wear of the cylinder and piston will take place, due largely to interference with proper lubrication. In proper quantities it probably has some tendency to loosen the carbon, and, in accordance with a theory which has frequently been advanced, it may result in bringing about a uniting of the nascent oxygen with the carbon,

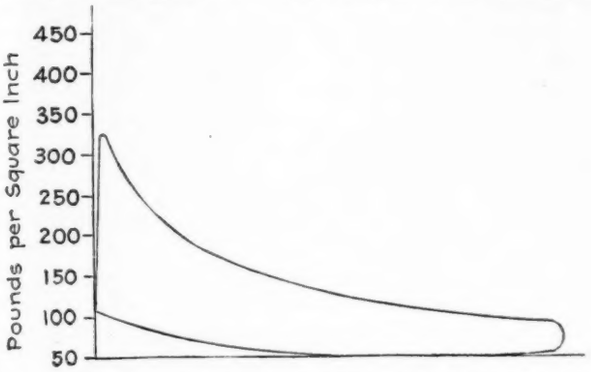


FIG. 4. DIAGRAM FROM TWO-STROKE-CYCLE VAPORIZING ENGINE AT RATED LOAD

thus keeping the cylinder slightly cleaner than would otherwise be the case.

LUBRICATION

Some years ago the auxiliary exhaust port in the four-stroke-cycle engine was considered good practice, and was extensively used. However, this port was later abandoned because it interfered with proper lubrication, and a dry streak through the cylinder was invariably found where these ports existed. The conditions in this respect are no different than in the two-stroke-cycle engines of today, although probably this port does not bring about so much excessive wear of the cylinder and piston as do the unburnt fuel and the water. Where crank-case compression is used, trouble with the lubrication of the main bearings is sometimes experienced.

Even with very short connecting-rods and with every available space in the crank case filled up so as to obtain as high a compression as possible, it is seldom possible to obtain a pressure in the crank case greater than 2¼ to 2½ lb. With the wear on the bearings, this pressure has a tendency to leak out and to interfere with their lubrication.

Leakage of crank-case compression also has a tendency to seriously affect the operation of the engine, as the transfer air is thereby lost and scavenging is not obtained.

This condition in conjunction with the filling up of the exhaust port with carbon, thus causing back pressure, has a tendency to equalize the pressures on both sides and interfere with proper scavenging. Incomplete scavenging means loss of power, wasted fuel, over-heated engine and premature ignition.

COMPRESSION

The compression employed by different manufacturers varies from 70 to 150 lb., some running as high as 300 lb. One manufacturer provides means of varying the compression in accordance with the fuel used and the conditions of operation. High compression should mean greater economy, but at the same time greater danger of excessive pressures. The amount of compression permissible will depend upon the temperature of the hot bulb, the temperature of the engine itself, the grade of fuel used, the method of injecting the fuel, and the time at which the fuel was injected. In consideration of these varying conditions, it is practically impossible to settle upon a satisfactory all-around compression.

Compression undoubtedly has greater influence upon economy than any other one point in connection with the design of internal-combustion engines. When the explosion engine was first attempted, no compression was used, and the engine was practically a failure on this account. Some fuels will ignite more readily than others, and, consequently, are more susceptible to heat of compression. Producer gas will stand a compression of from 150 to 160 lb., and natural gas 120 to 130 lb. without premature ignition. When liquid fuels are introduced into the cylinder of a four-stroke-cycle engine on the suction stroke, a compression of from 60 to 70 lb. is approximately all that can be obtained. Because of this low compression, an economy of less than 12,000 B.t.u. per b.hp.-hr. cannot be expected.

When high compression can be used, such as 120 lb. on natural gas, an economy of 8500 B.t.u. can be expected. This is the economy, therefore, which an engine of the pump-injection type should be able to obtain when using a compression of 150 lb. This economy is not attained, however, because the pump-injection principle is so far from perfect. In fact, it is seldom that an economy of 12,000 B.t.u. is had; the more common figure being 14,000 to 16,000 B.t.u.

FUELS

It is common for manufacturers of the hot-bulb type to state that any fuel can be handled in these engines. Most assuredly, any oil which contains heat units will vaporize when it comes in contact with a red-hot surface, and will consequently develop pressure and deliver power.

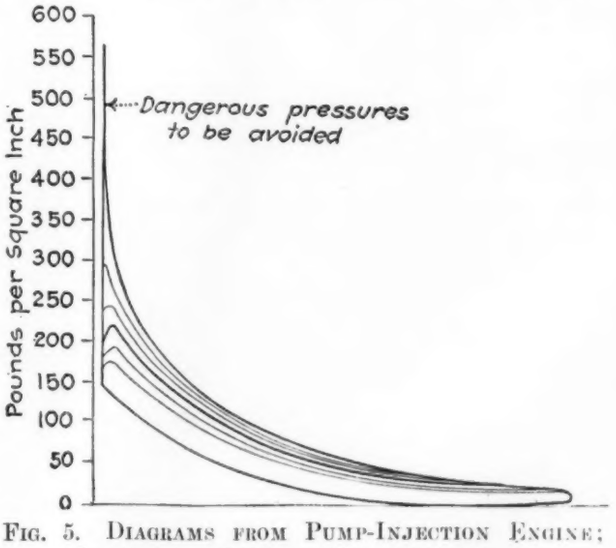


FIG. 5. DIAGRAMS FROM PUMP-INJECTION ENGINE; HEAVY LINE SHOWING EXPANSION CURVE AT RATED LOAD

This being the case, demonstrations can be made on very heavy fuels. Their continued success and satisfaction are, however, highly improbable, and in many cases lighter grades of fuel have been resorted to because less difficulty is experienced.

Can these engines operate successfully on kerosene? is a question which has been frequently asked. The ma-

jority of them do not, if they are built for operation on the heavier fuel. The flame propagation of kerosene is more rapid than with fuel oil or a heavier grade of distillate. This fact interferes with operation on either grade of fuel, as the flame propagation is dependent upon the temperature of the hot bulb, the timing of the injection of the fuel, the temperature of the cylinder walls, the amount of water injection used, and the load which the engine is called upon to handle. If these conditions are all correct for heavy fuels, they are not correct for lighter fuels, and except for a small range in variation, they are not adjustable.

It is true that all of the troubles mentioned do not exist in all installations of such engines. In fact, some of these engines operate without any of these difficulties, but they have all taken place in numerous instances, and any and all may take place in any installation.

Present State of This Type

The success achieved by heavy fuels in this type of engine is due to a peculiar combination of some of the following conditions: The high price of gasoline with the consequent desire to handle a heavier fuel; a consequent response on the part of some manufacturers to build an engine which they can claim will burn the heaviest liquid fuels; the fact that a certain vaporization of these heavier fuels takes place when brought in contact with extremely high temperatures; that there is available in some localities certain grades of heavy liquid fuels at a low price; that these engines are simple in construction and simple in appearance; that they do operate on these fuels and deliver power, and in some instances to the expressed satisfaction of the owner; and that this engine is passing through a stage at the present time when its shortcomings are being excused, and the owners are loath to admit that any difficulties are being experienced.

This attitude is due to lack of information regarding the actual causes of difficulties; consequently, there is a tendency to consider the troubles as due to improper operation or some outside influence. Excessive cylinder wear, for instance, can be laid to soft metal or to improper attention on the part of the operator; cracked bulbs and choked exhaust ports may be blamed on the operator or the fuel used; and blown-off cylinder heads or cracked beds may be attributed to insufficient water injection or possibly defective material.

Proper Fuels

Specific gravity indicates little as to the vaporizing qualities of a fuel, although the majority of fuels above 38 to 39 deg. gravity vaporize readily and the majority below 35 or 36 deg. gravity do not vaporize readily. The flash point also tells little, as fuels with a heavy body may contain sufficient lighter constituents to show a flash at a low temperature. A definite indication of the quality, however, is the boiling point, or the temperature at which different percentages of the fuel will distill.

Vaporizing Type of Engine

Another reason why the heavy-oil engine has been brought into favor is the numerous published statements that the so-called gasoline engine is not adapted to handling kerosene and the lighter distillates successfully. While this has unfortunately been proven in some cases, yet in others the application of lighter distillates to the four-stroke-cycle vaporizing type has been successful.

There are many features in connection with the proper design of the vaporizing type for handling the lighter grades of distillate and kerosene. There seems to be a mistaken impression that the carburetor is the essential feature. While the carburetor should be properly designed, there are other governing features equally important, such as piston speed, ratio of the stroke to bore, method of water circulation, location of the valves, shape of combustion space, location of igniter in combustion space, valve timing, voltage of ignition current, velocities through valves and intake passages, location of carburetor in relation to the inlet valve, contour of passages, governor valve and governor, method of automatically handling the mixture with varying loads, complete mechanical vaporization without resorting to preheating the charge, etc.

Engines Below 15 Horsepower

A large number of these small engines are used for farm work. They are usually of the four-stroke-cycle vaporizing type, using either gasoline or kerosene. The farmer should not attempt to use anything heavier than kerosene for this purpose. He should have an engine which is easily started, easily handled, and sure to run on a fuel which is readily obtained in small quantities.

It is doubtful if there is any other manufactured article which will vary to such a degree after it is assembled as will the gasoline engine. Pistons, piston rings and cylinders which are subjected to high temperatures and irregular distortion must be worn in together. Springs must be adjusted to the proper tension. Governors must be put in proper operating condition. Valve timing should be properly adjusted. Bearings should be worn in to avoid the possibility of their running hot. Brake tests should be made showing the engine developing its full rated horsepower. Water tests should be made of jacketed castings, as frequently leaks will develop after the engine has been in operation for some time. Moreover, the ignition system should be timed properly and checked up carefully. Considerable cost to the manufacturer can be saved by neglecting these matters, but the farmer should insist that the manufacturer produce an inspection sheet covering the detailed inspection and testing of the engine.

Conclusions

For many years after the internal-combustion engine came into general use, there was a current opinion that the engines were unreliable and tricky. This was brought about to a considerable extent by placing on the market a large number of engines which were not scientifically designed nor fully developed. It has taken many years of consistent effort to live down this prejudice.

We are now facing to a considerable extent a similar condition, because of the attempts to use heavy liquid fuels with a type of engine of cheap construction, such as has been heretofore described. Quite a large number of comparatively small manufacturers are experimenting with this proposition, with the result that undeveloped engines have been and are being placed on the market.

This type may in time achieve the results which it now claims, but in the meantime its failures should not be permitted to influence the industry as a whole. This can be accomplished best by a greater education of the public regarding the merits and demerits of the different types of engines, the different grades of fuel, and the purposes for which they can be used to the best advantage.

Small Marine Power Plant

By D. L. Roger

A small power plant, of interest because of its unique design and its economy in steam consumption, is shown in Figs. 1 and 2. This plant was installed in a 22-ft. launch and consists of a vertical boiler, vertical fore-and-aft compound engine and all necessary auxiliaries.

The engine has 3&6x4-in. cylinders and runs at a speed of 400 to 600 r.p.m. The boiler feed pump, air pump and fuel pump are driven by a drag crank and 2-to-1 gears from the forward end of the main shaft. The link on the low-pressure valve-gear is provided with an adjustment by which the receiver pressure is regulated. A reheater of U-shaped tubes connects the two cylinders.

The boiler is mounted aft of the engine. It is 12 in. high by 18 in. diameter and contains 426 half-inch tubes. The safety valve is set to blow at 200 lb. pressure. Kerosene is used as fuel in a special burner connected to a tank on which 70 lb. pressure is carried. The breeching carries the products of combustion through a superheater mounted on top of the boiler, around the high-pressure cylinder, through the reheater, around the low-pressure cylinder and to the atmosphere through a small stack mounted on the low-pressure cylinder.

Several tests have been made on the plant under working conditions on Lake Mendota, Wis. Numerous indicator diagrams have been taken of both cylinders and these show a steam distribution which is almost perfect. The engine consumes slightly less than 13 lb. of steam per brake horsepower-hour. The boiler, cylinders and breeching are so well insulated that it is possible to place the bare hand on any of these parts without discomfort. The gases escaping from the stack are exceptionally low in temperature.

Fig. 2. Power Plant in Launch

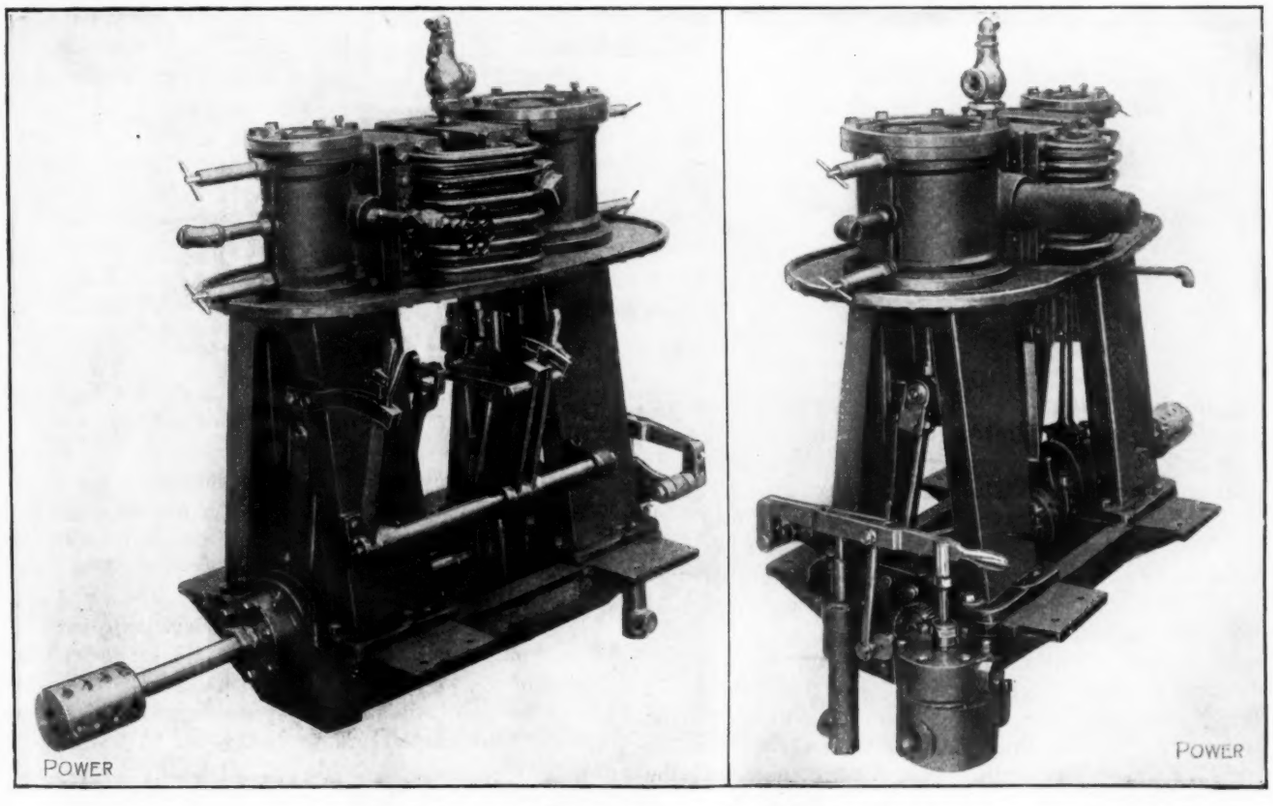


Fig. 1. Front and Rear of Compound Marine Engine

The steam from the boiler passes through a superheater, is expanded in the high-pressure cylinder, and is exhausted through the reheater to the low-pressure cylinder. After expanding in the low-pressure cylinder it exhausts into a keel condenser. The condensed water feeds the boiler. A filter is used through which the water is passed from the condenser overflow to the boiler-feed pump. The makeup water is also filtered.

J. C. White, chief engineer of the Capitol Power & Heat Co., Madison, Wis., designed and built this plant for his own pleasure. He started the work in 1909; the completed boat was put into the water in 1911.

Steam-Turbine Installation in Mexico

By John Klemm

For many years the Compañia Minera "Las Dos Estrellas" S. A., El Oro, Estado de Mexico, Mexico, has been purchasing its power from a public-service corporation. The slipping of one side of its greatest dam a few years ago seriously affected the power company for a time and caused heavy losses to innumerable customers. This company, being one of the several large consumers, in consequence installed a Westinghouse-Parsons multiple-expansion parallel-flow 1500-kw. steam turbine to generate three-phase current at 50 cycles and 3000 volts as a stand-by.

This turbine has a somewhat peculiar history. About two miles from the plant, in the center of a large masonry bridge, the trucks on one end of the car carrying the turbine broke down, precipitating the turbine to the river bed below, some 30 ft. Evidently, it did not wish to be installed at an 8000-ft. elevation. The machine was picked up uninjured except for a few paint scratches, and was installed and operated according to its specifications.

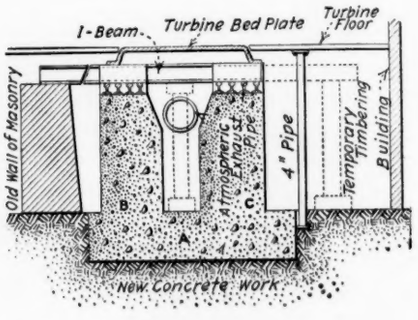


Fig. 1. Replacing a Turbine Foundation

Shortly after, the turbine wrecked itself so completely that an entire new spindle had to be purchased. Failure to start the auxiliary oil pump, unintelligent operation and weak masonry foundations, causing severe vibrations, were the causes, and conditions were made worse because of a 200-hp. compressor, operating on practically the same floor and foundation, whose vibrations caused a harmonic with the turbine vibrations. Another fault was that the I-beams were set directly on the masonry wall instead of being placed on some sort of material to distribute the weight; the I-beams thus sank into the masonry, and the machine, instead of resting on the entire length of the foundation wall, was practically where the I-beams had sunk.

At the time of the wreck, when the top half of the cylinder was raised someone remarked: "*Que buena ensalada*" ("What a fine salad"), adding the opinion that the most inefficient Corliss engine is far better than the most efficient steam turbine ever designed. This is a narrow-minded and prejudiced view, obviously untrue. The steam turbine has already proved its value, otherwise 30,000-kw. units would not be designed and operated, as they are today. Where intelligent and attentive labor is employed, there should be no chance for a steam turbine to wreck itself so completely as to require a new spindle. The turbine is not a very complicated piece of machinery; nevertheless, like any high-class apparatus it is not intended to stand abuse.

In this instance, to avoid further trouble it was decided to build a new foundation of concrete of the proportion 1 to 3 to 5, directly under the turbine, and do away with the masonry foundations, so designing the new work that the turbine would have absolutely no connection with the building or other apparatus, except the condenser, etc. The change was made with some difficulty, inasmuch as the machine had to be kept ready for immediate use in case of emergency.

From Fig. 1 an idea can be had of the conditions. The dotted lines show the temporary timbering. The space directly beneath the turbine, between the old masonry walls, was dug up and the concrete footing *A* laid. Wall *B* was built up to 5 in. below the 15-in. I-beam shown, and 12x12-in. and 8x8-in. timbering was placed between the new wall *B* and the old wall at *C*. This green timber was kept wet to prevent it from shrinking, so that the

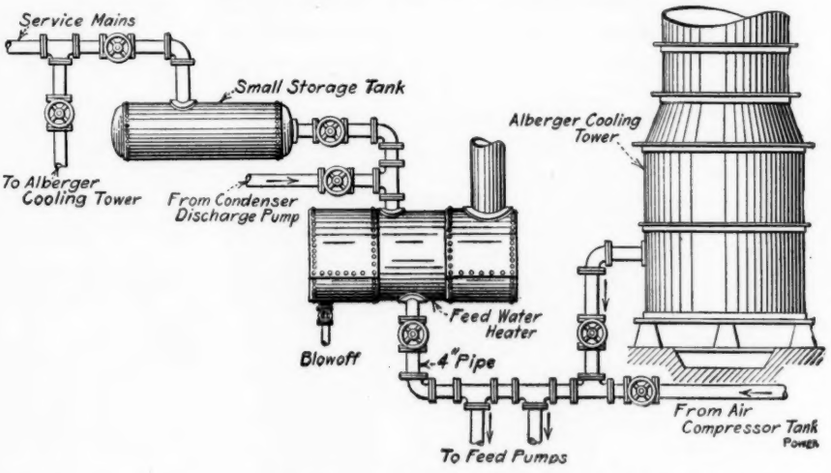


Fig. 2. Old Boiler Used as a Feed-Water Heater

weight of the turbine could not come on the condenser. The old wall at *C* was then removed, a new concrete footing laid and the wall *C* built up and steamed to crystallize the cement, which was done in about 60 hr., thus avoiding the slow 28-day process.

For the steaming ½-in. pipe, previously drilled with ⅛-in. holes about 6 in. apart, was laid all around the bottom of the wall; the entire wall was then well covered with a heavy canvas and the steam turned on. At the beginning the temperature was held below 35 deg. C., then gradually increased to 70 deg. C., at which it was maintained for the last 12 hr. Great care must be taken not to overheat the concrete and thus ruin the entire work, especially if it is reinforced, as the coefficients of expansion for iron and concrete are different. The steam pressure never exceeded 1.5 atmospheres, and with this it was found that the best work was done on the top of the wall. After the steaming the electrical end was dried out before being placed in service, to avoid danger of a burnout.

Full-length 40-lb. steel rails were then placed on the new walls *B* and *C*, Fig. 1, and 1-in. plates and steel wedges set tight against the 15-in. I-beams on which

rested the turbine bedplate. These rails served to distribute the weight of the machine over the entire length of the walls. All spaces between and around the rails and I-beams were then filled in with concrete to the level of the bedplate, making of the whole almost a solid box-like construction with the opening in the center. It may be criticized as inconsistent with uptodate construction not to allow more space below the unit, but this could not well be avoided, under the existing conditions; moreover, the space was sufficient, as all high-speed machines, especially turbo-generators, have artificial ventilation, cold air being blown into the windings. Without this precaution the air in the machine while it is running would be churned and would seriously affect the temperature rise.

The timbering was finally removed and the I-beams, one end of each of which was still in the old wall, were cut in a diagonal direction, as shown by Fig. 1, so that in settling, the I-beams could not again make contact with the old work, the tendency being to settle away.

An additional installation of three 400-hp. Babcock & Wilcox water-tube boilers brought the total boiler capacity up to 2400 hp. This addition may seem unnecessary, and, hence, a useless expenditure, but steam is always required for purposes other than power generation, and it is desirable to have ample margin for cleaning, etc. Further, it does away with the need for induced or forced draft and attendant loss of heat up the chimney, the extra cost of the apparatus and the disadvantages of boilers.

There are no peaks to consider; the load is steady throughout the 24 hr. The lighting load at night is hardly perceptible. The boiler feed water is taken almost directly from the condenser, a 4-in. pipe being tapped into the 12-in. discharge pipe of the circulating condenser pump (see Fig. 2). This 4-in. pipe discharges into an old fire-tube boiler used as a heater, and the water is then brought up to 90 deg. C. from the heater. It flows by gravity to the feed pumps, of which there are two Worthington 10x6x10-in. The pumps are also connected to the Alberger cooling tower and a tank, which is used to cool a 200-hp. Ingersoll-Rand compressor, so when the turbine is not in operation the boilers still get warm water. Steam is required for other purposes for 24 hr. daily, and is held for emergency. The heater is also connected to the company's service mains through a small storage tank above the heater, as Fig. 2 shows.

There are two wrought-iron expansion joints in the 8-in. steam header line, one U between what are known as No. 4 and No. 5 boilers, and a 90-deg. elbow between No. 1 boiler and the turbine. These suffice to take up any expansion in the header.

American fuel shipped to these regions comes high, costing on an average about $24.50, Mexican currency, per ton of 1000 kg. (2205 lb.) on the plant grounds.

The generator room is 39 ft. 8 in. by 89 ft., and is constructed of masonry. The doors and windows have a brick facing, presenting a neat and attractive appearance. The boiler room is 39 ft. 8 in. by 92 ft., and is constructed of the same class of material.

Coal Storage Tests—In his annual report as Chief of the Bureau of Steam Engineering, Admiral R. S. Griffin says: "The coal stored at New London under the three different conditions, in the open, under cover, and under water, was given the third annual evaporative test. No marked difference in evaporative efficiency was shown between the coal stored under different conditions, and no conclusive evidence developed as to the best method of storing coal."

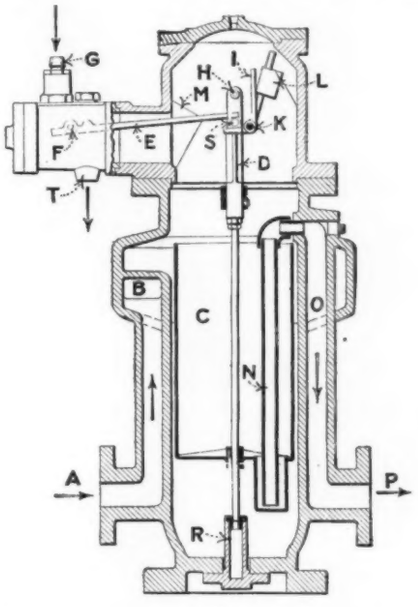
New Return Steam Trap

This trap is of the closed type, with the moving parts arranged inside the apparatus. It operates automatically when sufficient condensate has accumulated to move the float. It has few moving parts; therefore friction and wear are reduced to a minimum, and there are no stuffing-boxes.

The trap is made with a cast-iron body, brass inside parts and an open copper float. It is built for working pressures up to 180 lb. For higher pressures the body is made of steel.

In operation the condensate enters the trap at *A* and flows through opening *B* into the body. The water level inside of the apparatus rises until it reaches the top of the open float *C*, which is shown in its highest position. The float fills, loses its buoyancy and starts to sink, carrying with it the stem *D*, which pulls down on lever *E*. This lever, turning on the center *F*, opens the steam valve *G*.

As the float continues to sink, the roller *H* comes in contact with the short arm of the lever *I*, which, turning



Section through Return Steam Trap

about the center *K*, throws the attached counterweight *L* over to the left until the long arm of the lever presses on the roller *H*. The weight, falling over to *M*, will cause the roller, which is already in contact with the steam-valve lever *E*, to drop quickly. This gives a sudden increase in the opening of the steam valve, and the full pressure of the steam acts on the water in the trap, discharging it through the pipe *N*, the chamber *O* and the outlet *P*. A dashpot *R* takes up the shock caused by the counterweight falling on *M*.

A check valve on the inlet *A* prevents water from entering the intake pipes, and another in the discharge line prevents the return of the discharged water.

The empty float rising, throws the counterweight back to its original position and releases the lever *E*, allowing the steam valve to close. As the float continues to rise, the roller *S* lifts the lever *E* and opens the valve *T*, thus relieving the pressure in the trap. The steam which flows through this valve is received in a tank and condensed.

This trap is manufactured by the General Condenser Co., 1240 North 12th St., Philadelphia, Penn.

Why Direct-Current Motors Fail to Start--I

By F. A. Annett

SYNOPSIS—Directions for systematically locating the trouble, should a motor fail to start when the starting-box handle is thrown over.

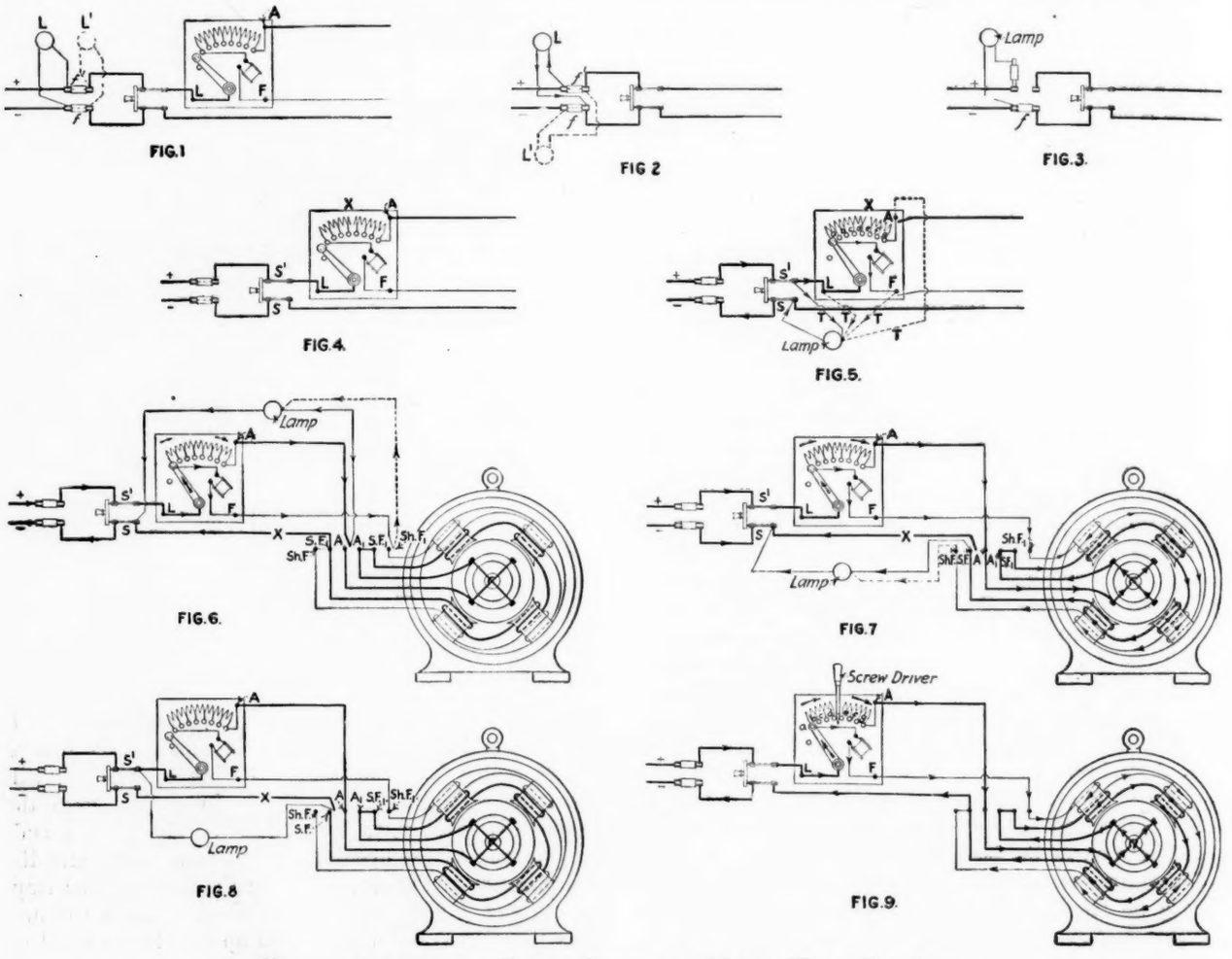
When locating trouble in a motor, it is essential that certain features be given first consideration. Assume a case where the motor fails to start because of a blown fuse (*f'*, Fig. 1). The first thing to ascertain in all such cases would be if the line is alive. This can be determined by connecting a lamp or voltmeter as at *L*. If the lamp lights, next test below the fuses as at *L'*. In this case it will not light, for the fuse *f'* is blown.

Most inclosed fuses indicate when they are blown, but this cannot always be relied upon, and when locating trouble the best policy is to depend upon nothing but what has been determined by test. To test the fuses without removing them, connect the lamp as illustrated in Fig. 2. If the lamp is connected as at *L*, it will light, indicating that fuse *f* is not defective. When connected as at *L'*, it will not light, for the circuit is open in fuse *f'*, which may be removed and again tested, to make sure that a mistake has not been made in the test. This can be done as indicated in Fig. 3. If the fuse is blown, the lamp will not light.

If link fuses are used, the fact that they look good should not be taken as final, for a fuse may be broken off close to one of the terminals, and although it may look good on inspection, it is nevertheless open. Another point that should not be overlooked is the switch. One of the clips may be worn or sprung just enough to prevent it from touching the blade. Therefore, if the fuses test good, do not neglect to test the switch and see that the clips are making good contact and that none of the connecting wires are broken at the switch terminals.

If the break is in some part of the circuit other than the fuse, such as at *X* in the starting resistance, Fig. 4, an indication that the circuit is alive will be given when the starting arm is brought up to the first or second contact and allowed to drop back to the off position; that is, a spark will occur when the field circuit is opened.

If the motor does not start on the first or second con-



How to Locate the Fault When the Motor Will Not Start

tact point, bring the arm back to the off position and look for the cause. If it is loaded, first determine whether the load is free so that it can be started; also, that the motor bearings are not set on the armature shaft or worn so as to allow the armature to rub against the polepieces. The writer recalls an instance in which he was called in to repair a pump motor that another electrician had been working on all day, trying to get it to run. Upon attempting to turn the pump over by hand, it would not move, and the cause was traced to freezing of the pipe line running to the roof tank. There was nothing wrong with the motor or controller except what was caused by the ordeal they had been put through during the day.

After it has been determined that everything is favorable for the motor to run, the next thing is to make an inspection of all the electrical connections to see that they are tight and making good contact and that none of the wires are broken off at the connections; for sometimes a wire breaks off and will open only enough to interrupt the circuit, and unless it is moved it cannot be detected. If lugs are used on the wires, see that they are properly soldered, for if this has not been thoroughly done the wire will corrode in the connection and may cause an open circuit. All this is a hard and fast rule which may be applied to any motor whether direct- or alternating-current. Furthermore, it is well to make sure that the brushes are making good contact on the commutator and are free in the brush-holder pockets.

If the foregoing has failed to disclose the reason for the motor not starting, next test for purely electrical troubles. In this connection, first test the starting device. Fig. 5 shows a convenient way of doing this. Disconnect the armature and field connections on the starting-box and place the arm on the first contact, as shown. Then connect one lead of the test lamp to the switch terminal S (the one connecting directly to the motor), and to make sure that everything is in condition to make the test, connect the other lead T of the lamp to the other switch terminal S'. If the proper indication is had, next connect lead T to the "L" (line) connection on the starting-box. If the lamp continues to light, connect lead T to terminals A and F, as indicated, which in the present case will show a complete circuit through the starting-box to the terminal F, as indicated by the arrowheads, but not to terminal A, for the resistance is open at X. That is, when connected to F the lamp will light, but not when connected to A.

The exact location of the fault can be easily determined by testing to the contacts on the resistance. If lead T of the lamp is connected to contacts e, f or g, which are to the right of the break in the resistance, the lamp will not light, but when connected to d it will light. Since the lamp lights at d and not at e, it indicates that the circuit is open between these contacts.

A quick way to repair this fault is to drive a piece of fuse wire in between the contacts on the front of the slate between which the fault is located. A better and more permanent way is to remove the cover from the starting-box and repair the break in the resistance coil. If this cannot be located, as in some cases the coils are molded into a compound, the two contacts can be connected together on the back of the slate with a piece of wire.

In Fig. 6, X shows a break in the wire which connects the series and shunt fields direct to the switch. If the tests previously described are made and the circuit through the starting-box has been found complete, the next step will be to disconnect the two connecting wires between the motor and the starting-box, at the machine, and test through them as indicated. This will show a closed circuit, as represented by the arrowheads. Next connect the armature and field wires to their respective terminals and test through the armature and field coils, as in Fig. 7, and if they are not defective the lamps should light.

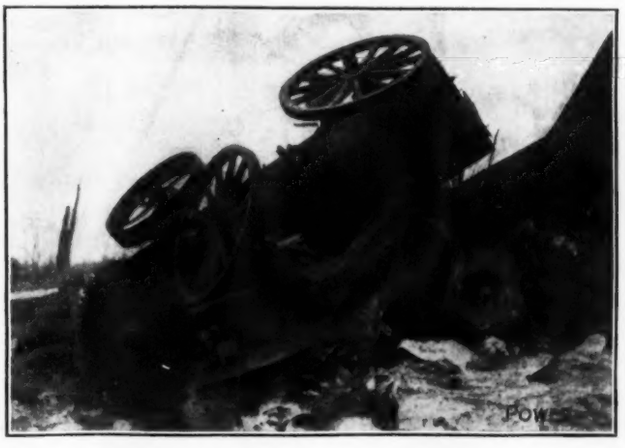
There is but one thing left to test and that is the connecting wire from the switch to the series and shunt field connections. To do this, connect one lead of the lamp to the switch terminal S' and the other to the end of the wire at the motor, as indicated in Fig. 8. In this case it will not light, which indicates that the circuit is open between the switch and the other end of the wire. The defect may then be definitely located and repaired, or the wire replaced by a new conductor. If the wires are in conduit or molding, the defective one should be replaced by a new wire, for a spliced wire in molding or conduit is a violation of the Board of Fire Underwriters' rules.

A practical way of testing for an open circuit in the starting resistance is illustrated in Fig. 9. Close the switch and bring the starting-box arm upon the first contact, and then bridge between the contact buttons with a piece of metal; a screwdriver being convenient for this purpose. If the break is in the starting resistance, such as between d and e, the motor will start when the defect is remedied; then the switch may be opened and the fault repaired, as previously explained.

An open circuit in the series field may be located as described in "Testing for Open Circuits in Field Coils," (POWER, Aug. 4, 1914).

Portable Boiler Exploded

This old boiler which the owner had recently purchased was being given a "tryout" on a wood saw. The engineer said the engine ran "snappy" with the throttle valve two



ENGINE'S POSITION AFTER EXPLOSION

turns open, which is taken as indicating that there was an excessively high pressure in the boiler, although the steam gage showed only 80 lb. and the safety valve was set, by the same gage, to blow at 100 lb.

The failure occurred at the bottom of the firebox, of the water-bottom type, where there was a section approximately 26 by 30 in. without stay-bolts. This section was forced upward, throwing the grate bars out through the fire-door. One section in its flight struck a dinner pail carried by a schoolboy, who was passing at a distance of about 200 ft., tearing it from the bail. The force of the explosion impelled the engine forward, nearly overturning it on the woodpile, as shown in the illustration. Two men were slightly injured.

Waste-Heat Boilers*

By Osborn Monnett†

SYNOPSIS—Smoke prevention in typical metallurgical furnaces operated in connection with waste-heat boilers. The latter are provided with independent furnaces.

Quite frequently the hot gases from metallurgical furnaces are available for steam making. When boilers are combined with such furnaces it is sometimes desirable to so arrange them that they may be fired by hand when the furnace is down. In this case provision for smokeless operation can be made by installing one of the hand-fired furnaces shown in previous articles. Care must be exercised in selecting a furnace adaptable to the particular type of boiler being used.

A typical installation is shown in Fig. 1. This boiler receives the waste gases from a billet-heating furnace, the gases coming through the perforated side walls of the setting. When this furnace is not in use the boiler can be fired by hand in the usual manner. As shown, the setting consists of a standard tile-roof furnace, with deflection arch, siphon steam jets and panel doors. Many metallurgical furnaces have underground breechings and it is generally simple to lead the gases to the boiler.

The underfeed type of stoker is especially adapted for the smokeless operation of metallurgical furnaces because it does not depend on natural draft for its air supply.

*Copyright, 1915, by Osborn Monnett.

†Smoke Inspector, City of Chicago.

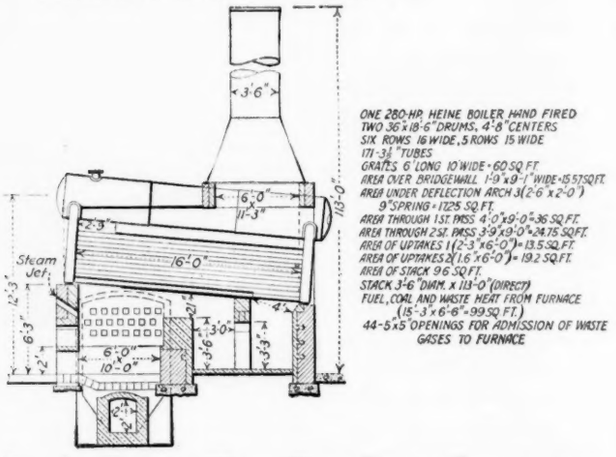

ONE 280-HP. HEINE BOILER, HAND FIRED
TWO 36"x18'-6" DRUMS, 4'-8" CENTERS
SIX ROWS 16 WIDE, 5 ROWS 15 WIDE
171-3½" TUBES
GRATES 6' LONG 10' WIDE = 60 SQ. FT.
AREA OVER BRIDGEWALL 1'-9"x9'-1" WIDE = 15.5 SQ. FT.
AREA UNDER DEFLECTION ARCH 3(2'-6"x2'-0")
9" SPRING = 112.5 SQ. FT.
AREA THROUGH 1ST. PASS 4'-0"x9'-0" = 36 SQ. FT.
AREA THROUGH 2ST. PASS 3'-9"x9'-0" = 24.75 SQ. FT.
AREA OF UPTAKES 1(2'-3"x6'-0") = 13.5 SQ. FT.
AREA OF UPTAKES 2(1.6'x6'-0") = 19.2 SQ. FT.
AREA OF STACK 9.6 SQ. FT.
STACK 3'-6" DIAM. x 113'-0" (DIRECT)
FUEL, COAL AND WASTE HEAT FROM FURNACE
(15'-3"x6'-6" = 99 SQ. FT.)
44-5"x5" OPENINGS FOR ADMISSION OF WASTE GASES TO FURNACE
Steam Jet
3'-6"
16'-0"
11'-3"
113'-0"

Fig. 1. Typical Waste-Heat Boiler Setting with Auxiliary Hand-Fired Furnace

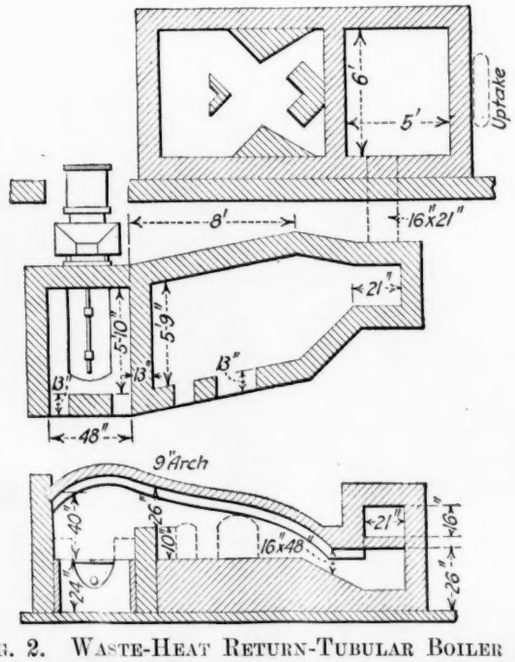


Fig. 2. Waste-Heat Return-Tubular Boiler and Forging Furnace with Underfeed Stoker

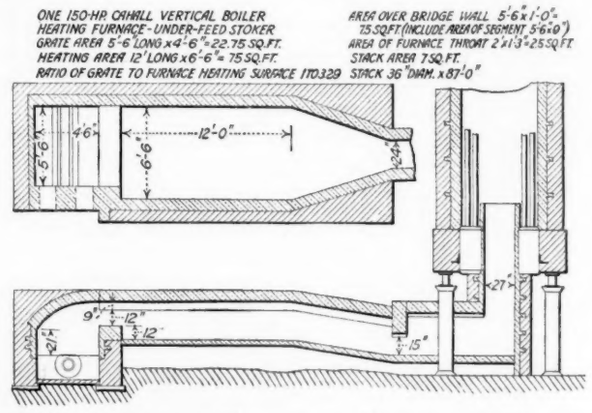

ONE 150-HP. CAHALL VERTICAL BOILER
HEATING FURNACE - UNDER-FEED STOKER
GRATE AREA 5'-6" LONG x 4'-6" = 22.75 SQ. FT.
HEATING AREA 12' LONG x 6'-6" = 75 SQ. FT.
RATIO OF GRATE TO FURNACE HEATING SURFACE 1 TO 3.29
AREA OVER BRIDGE WALL 5'-6"x1'-0" = 7.5 SQ. FT. (INCLUDE AREA OF SEGMENT 5'-6"x0")
AREA OF FURNACE THROAT 2'x1'-3" = 2.5 SQ. FT.
STACK AREA 7 SQ. FT.
STACK 36" DIAM. x 87'-0"
12'-0"
6'-6"
4'-6"
27"
9"
12"
15"

Fig. 3. American Underfeed Stoker and Heating Furnace Connected to 150-Hp. Cahall Vertical Waste-Heat Boiler

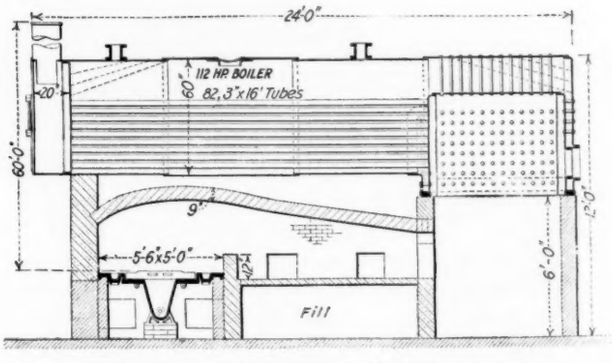


Fig. 4. American Underfeed Stoker and Forging Furnace with 112-Hp. Firebox Boiler

Owing to the length of the gas passes and number of turns, etc., there is generally but little suction over the fire from natural draft. Any stoker depending on natural draft and an ignition arch is at a big disadvantage from the standpoint of maintenance and capacity. An underfeed stoker will operate satisfactorily under conditions where a natural-draft stoker would burn up from the heat bottled in the furnace.

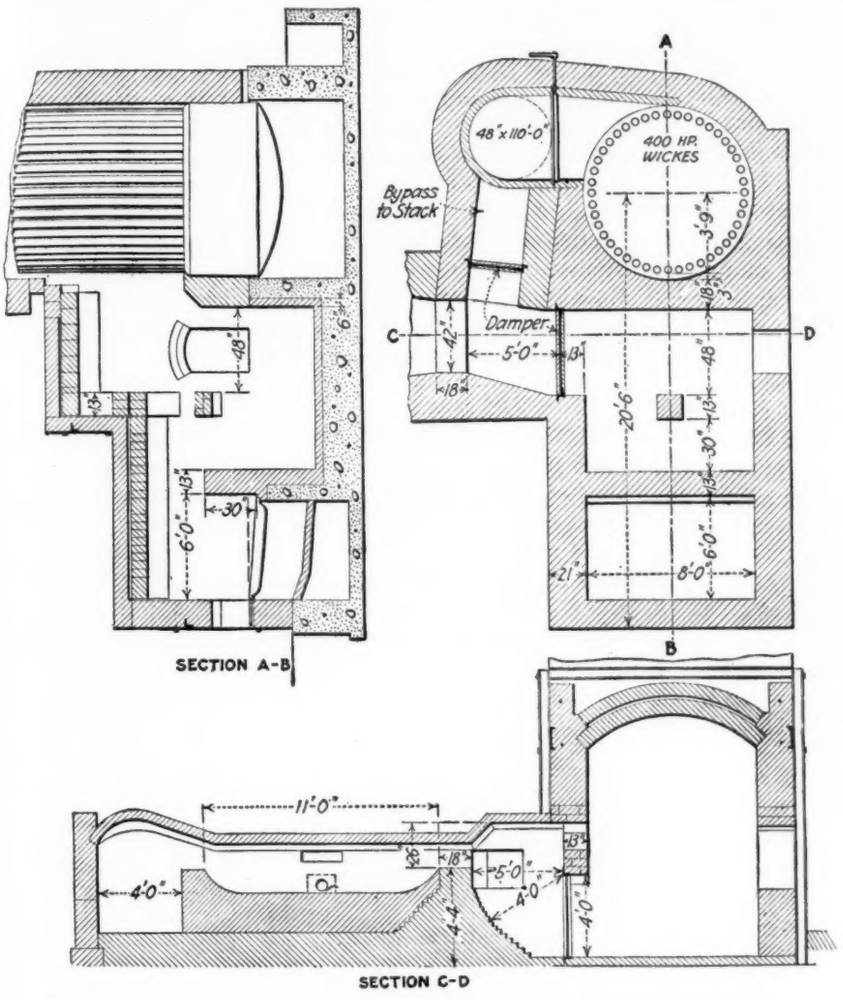


FIG. 5. JONES UNDERFEED STOKER AND MALLEABLE-IRON MELTING FURNACE WITH 400-HP. WICKES WASTE-HEAT BOILER

Fig. 2 shows a waste-heat boiler of the horizontal return-tubular type installed in connection with a reheating furnace having an underfeed stoker. The boiler itself is equipped for hand firing, an ordinary No. 8 furnace containing a pier and wing walls being installed behind the bridge-wall. The usual rules for furnace areas based on the grate surface are allowed. If the boiler is to be used much, independently of the waste-heat furnace, the same stack height required by an independent boiler should be provided.

Another combination of metallurgical furnace and waste-heat boiler is shown in Fig. 3. In this case the ratio of grate to metallurgical heating surface is given as 1 to 3.29. The gases pass up through the lower drum of a specially constructed Cahall vertical boiler. Obviously, the proportions of the furnace depend on the product to be heated, so that the design must be varied to suit the conditions. The accompanying illustrations merely indicate how the different combinations can be operated without smoke.

Forging furnaces are often connected to waste-heat boilers, as the steam raised by the boilers can be used to advantage in the steam hammers, the same fuel sufficing for all operations in the shop. Fig. 4 shows an interesting and compact installation of this kind, consisting of a 112-hp. firebox boiler over a forging furnace fitted with an underfeed stoker.

When hand fired, the malleable-iron melting furnace produces a great deal of smoke. The underfeed stoker can be applied to this furnace with advantage and the waste heat used for steam making, as shown in Fig. 5. The latter consists of a 400-hp. Wickes waste-heat water-tube boiler attached to a melting furnace using an underfeed stoker and having an auxiliary hand-fired furnace for emergency purposes. With a bypass to the stack the melting furnace can be operated when the boiler is down for cleaning or repairs. Also, the furnace can be cut off from the boiler and the latter operated independently by closing the firebrick curtain-wall or damper, as indicated in the drawing. This makes a flexible combination. Another forging furnace with a Jones self-cleaning underfeed stoker is shown in Fig. 6, attached to a Wickes waste-heat boiler.

Belt-Driven Coal Crushers—The Edison Electric Illuminating Co., Brooklyn, N. Y., has found it advantageous to sacrifice plant efficiency obtained by using steam engines and to use motor drive with belt transmission for driving its coal crushers, as an obstruction in the crusher will either throw the belt or open the motor circuit-breaker instead of breaking parts of the crusher. Car couplings and pieces of steel frequently passed into the crusher with the coal, which stopped the rolls and the engine with a shock, thus subjecting the parts to excessive stresses. With belt drive the crusher and motor are protected against injury.

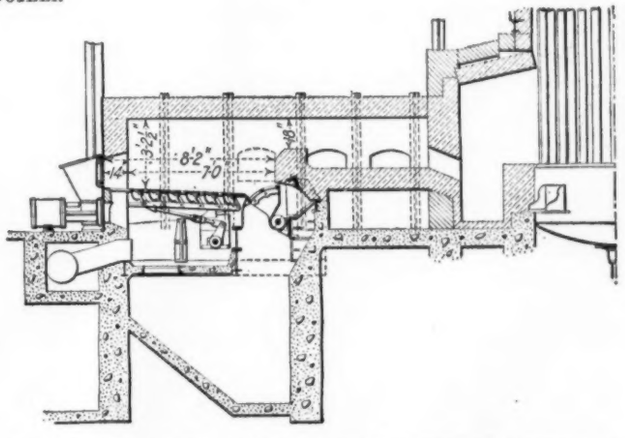


FIG. 6. JONES SELF-CLEANING UNDERFEED STOKER IN FORGING FURNACE AND WICKES WASTE-HEAT BOILER

Selecting a Pump for General Service

By Charles L. Hubbard

SYNOPSIS—The article takes up briefly some of the more important points to be considered in the selection of a pump for general service in connection with power and industrial plants.

When the water is taken from the public mains or flows to the plant by gravity, the problem is simple and usually involves only the proportioning of pipes to the pressure and volume required. When the power house is at a higher elevation than the source of supply, the water must be pumped. If the conditions are such that the friction head in the suction pipe, plus the elevation, does not exceed fifteen to eighteen feet, the pumping equipment may be placed in the power house; otherwise, it must be located at some intermediate point where this limit will not be exceeded. Direct-acting steam pumps, engine- and turbine-driven plunger pumps and centrifugal pumps are adapted to the first of these conditions, and also to the last when the distance is not so great as to make the carrying of steam from the power house both expensive and wasteful. When the distance exceeds a certain limit it is usually better to drive the pump by an electric motor or gasoline engine than to install and care for a special boiler.

When the water is taken from a river, and the grades are suitable, a hydraulic ram may be employed where there is an abundance of water. This device is made in a large number of sizes and requires practically no attention, as the only parts subject to wear are the rubber valve-disks. With artesian wells there are two methods in common use for pumping the water. In the first, each well is equipped with a lift-pump driven by steam, electricity or gasoline, and connected with a common main leading to the power house. The second method makes use of the "air lift" and is especially adapted to cases where it is desirable to increase the flow and to plants using a series of wells, as one compressing outfit in the power house may be made to do the entire work.

Direct-Acting Steam Pump

This type is made in a variety of forms and sizes and is widely used for power-plant work. Piston pumps are adapted to locations where the water is free from grit or other substances likely to destroy the packing. When these are present, the plunger pump is preferable on account of the ease with which the worn parts may be repacked or renewed. One of the chief disadvantages of the direct-acting pump is its excessive steam consumption as compared with an engine or turbine, but this is offset in many cases by the low cost of installation, convenience and ease with which the speed may be regulated to meet varying requirements. Pumps of this type are made single, duplex, simple and compound, according to requirements.

Direct-acting pumps have an average mechanical efficiency of 65 to 75 per cent. and a "slippage" of 15 to 20 per cent. under ordinary conditions of adjustment. The steam consumption of small and medium duplex pumps will run from 80 to 160 lb. per developed horsepower, per hour, according to the size. By compounding, this may be reduced from 40 to 50 per cent. Pumps of this type are operated at a comparatively low speed, although the steam consumption per unit of work decreases as the speed increases. For large sizes the piston speed is usually limited to 100 ft. per minute, but for strokes of less than twelve or fourteen inches, the piston speed should be reduced proportionately. Pumps which are to run continuously should be designed to operate at about one-half the maximum allowable speed noted above.

Power Pumps

Power or geared pumps are used for practically the same purposes as the steam pumps just mentioned, but they are more economical to operate as they may be driven by an engine, turbine or motor. When belted to line shafting or driven by prime movers requiring a constant speed, they are not so desirable as the steam pump, owing to the difficulty of regulation. When used for supplying tanks and reservoirs or other purposes where they may be run at constant speed for long periods, they give satisfactory results and are supplanting the steam pump in many lines of service.

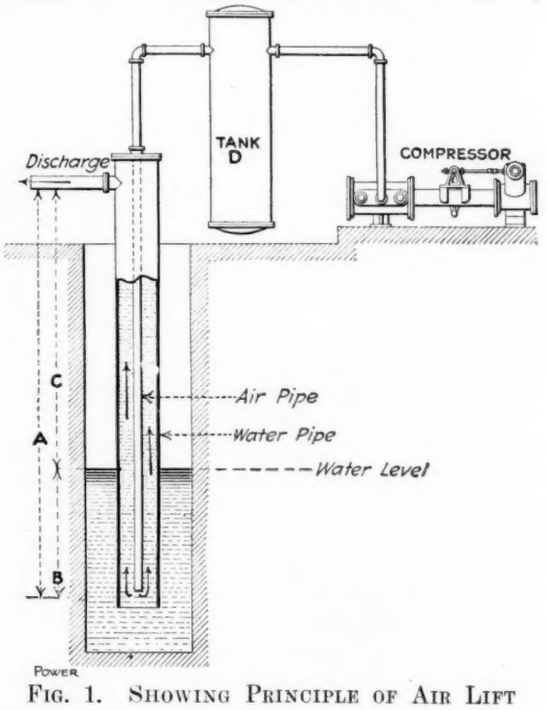


Fig. 1. Showing Principle of Air Lift

The efficiency of the triplex pump may be taken as about 60 per cent. for total heads of 100 ft., 70 per cent. for 200 ft., and 80 per cent. for 300 ft. The slippage is usually from 15 to 20 per cent.

Centrifugal Pumps

Pumps of this type have come into general use with the advent of the electric motor and the steam turbine. These are of two general forms, the "volute" and the "tur-

bine," varying chiefly with the interior construction of the casing.

The volute pump is usually single-stage, and limited to heads of 100 to 120 ft., although two-stage machines are constructed for much higher pressures. Turbine pumps are designed for high lifts and are usually compounded in order to reduce the peripheral velocity and thus reduce the friction. It is important when using a centrifugal pump of any type to select one designed for the conditions under which it is to operate.

The efficiency of centrifugal pumps commonly runs from 60 to 80 per cent. for the better types, working under the conditions for which they were designed. The slippage varies from about 20 to 60 per cent., according to size and construction.

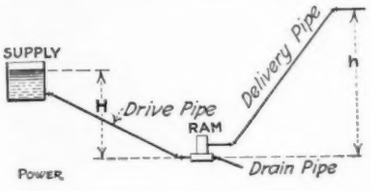


FIG. 2.

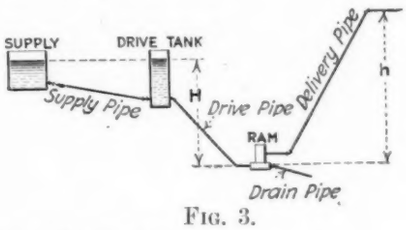


FIG. 3.

METHODS OF CONNECTING HYDRAULIC RAMS

Among the advantages of this pump are simplicity and compactness, absence of valves, low cost, uniform delivery and high rotative speed, adapting it to direct connection with motors and turbines. On the other hand, it is not possible to obtain as high an efficiency as with the best designs of piston pumps when the latter are kept in first-class condition. Furthermore, the speed cannot be varied, except within narrow limits, without loss of efficiency.

DEEP-WELL PUMPS

Deep wells are of two kinds—open wells having a large diameter, and driven or artesian wells. The type of equipment required in the first case consists of one or more pump cylinders placed within eighteen or twenty feet of the surface of the water and connected with some form of pump head at the top of the well by means of a long rod. The water is raised to the cylinder by suction and is then lifted or forced from this point to the surface of the ground.

With an artesian well an outer tube is driven to the required depth, extending to the surface of the ground. Inside of this, submerged in the water near the bottom, is the "barrel" containing the pump bucket and foot valve. The bucket or plunger is connected with a pump head at the top of the well by means of a wooden sucker rod, this material being used in order to reduce the weight. Pumps of this kind may be operated by a direct-acting steam cylinder or by a geared electric motor or a gasoline engine.

Deep-well pumps have an efficiency of 40 to 50 per cent. and a slippage of 10 to 15 per cent.

AIR LIFT

The principle of the air lift is shown in Fig. 1. A water pipe is carried down to the required depth, together with an air pipe either on the outside or the inside, as convenient. Compressed air is forced into the water pipe near the bottom, thus decreasing the density of the water within it, due to the air bubbles, and an upward flow is produced by the difference in weight between the column of solid water and the mixture of air and water. Air under sufficient pressure for raising the water is furnished by a compressor in the power house, the air pipe following the line of the water pipe.

The tank D is for equalizing the pressure and reducing the pulsation between the strokes. The distance B is called the "submergence," C the lift, and A the total head. In practical work the submergence is expressed as a percentage of the total head. For example, if A and B are 250 and 150 ft., respectively, the submergence is

$$\frac{150}{250} = 0.60$$

or 60 per cent. The efficiency of an air lift increases with the percentage of submergence and commonly runs from about 30 per cent. for

$$\frac{B}{A} = 0.5$$

up to 50 per cent. for

$$\frac{B}{A} = 0.8$$

A ratio of about 1 to 6 between the areas of the air and water pipes gives the best results for average conditions. If the air pipe is too large, power will be wasted in a high water velocity, and if too small, the air bubbles will not expand sufficiently to fill the discharge pipe, but will rise through the water without lifting it.

HYDRAULIC RAM

This offers the cheapest means of pumping where there is a sufficient supply of water and suitable grades.

Two general methods of connecting a hydraulic ram are shown in Figs. 2 and 3. If the drive pipe is too long, the excessive friction will interfere with the proper action of the ram, and if too short, water will be forced back into the drive tank. In practice it is customary to make the length of drive pipe equal approximately to the lift (h) to the tank or reservoir. When it is necessary, for any reason, to locate the ram at a greater distance from the source of supply, the required length of drive pipe can be secured by introducing a standpipe or intermediate drive tank nearer the ram, as shown in Fig. 3. For large quantities of water the fall from the source to the ram should not be less than two feet and, unless special provisions are made, should not in general exceed twelve or fifteen feet, owing to the shock when the flow is suddenly checked in the drive pipe.

Standard rams are made in large sizes, using from 400 to 15,000 gal. of water per minute (G), operating under a fall of 1½ to 50 ft., and raising water 35 ft. per foot of fall, up to a maximum of about 800 ft.

The working formulas for the hydraulic ram are as follows:

$$g = \frac{2 \times G \times H}{3 \times h}, \qquad G = \frac{3 \times h \times g}{2 \times H}$$

$$H = \frac{3 \times h \times G}{2 \times G}, \qquad h = \frac{2 \times G \times H}{3 \times g}$$

in which

g = Gallons discharged by ram;
G = Gallons required for operating the ram;
H = Fall, in feet, from source of supply to ram;
h = Height, in feet, to which water is lifted above the ram.

Chart for Coal Purchasers

By W. V. Bowles

Although coal is bought for evaporating water, few buy it on an evaporative basis. The reason usually given is that the human element or "error" cannot be accounted or compensated for. Most engineers agree that the evaporative basis is the correct one on which to buy coal.

Assume a plant which requires, say, 3000 or more tons of coal a year, and it is desired to purchase coal on an evaporative basis. The first thing to do is to run a test on the boilers with various coals and make a chart from the records obtained. The accompanying chart is plotted from a stoker plant and illustrates the idea and plan of procedure.

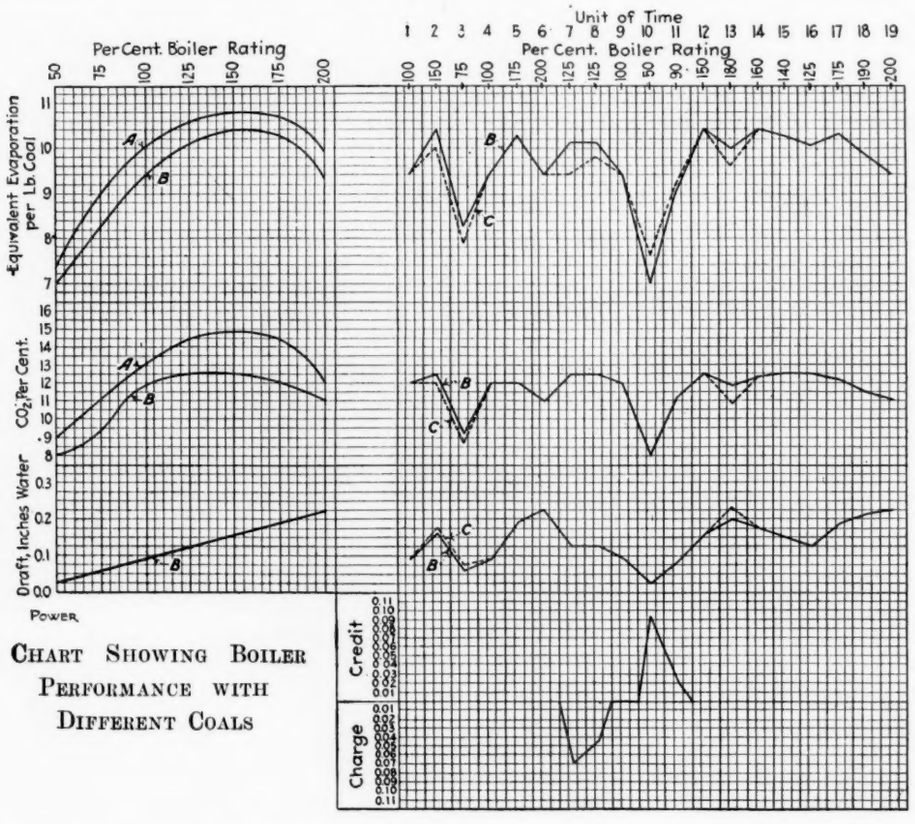


Chart Showing Boiler Performance with Different Coals

It is understood that the coal company should have a competent representative present during the test and at the calibration of all instruments and to have access to all records. First, the attainable evaporation and CO_2 curves are plotted at all boiler loads, together with a curve showing the best draft to use. Below is plotted a curve showing what is in the plant, termed "the adopted standard." This is based on a longer test when uncontrollable conditions may be taken into account.

The chart is also divided into a number of time periods, hours or days, as may be found most suitable. As each period is ended the average boiler rating is marked at the top, and below is plotted the adopted standard.

Referring to the charts under period 2, it is found that the average rating was 150 per cent. The equivalent evaporation should be 10.4 lb. per pound of coal, but the actual evaporation fell off to 10 lb. Following down the column, may be found the reason. The CO_2 dropped from 12½ to 12 per cent. Following still further down the column, the draft has increased from 0.16 to 0.175 in., showing the operating force to be at fault. A similar condition is indicated in the next column.

Under period 7 the evaporation dropped 0.069 per cent., but the operating conditions were according to adopted standards, when the coal company is penalized an equivalent of 0.069 on the coal burned during that period. If a better evaporation is gained than is shown by the standard line, as in period 10, the coal company is given a bonus equal to the increase; in this case, 0.092 per cent. If a shipment of bad coal comes in and the operating conditions are allowed to deviate from the standard, both will be shown in their correct proportion.

It may be found desirable to add draft lines to show conditions in other passes of the boiler, also temperature curves from the various passes. With proper checks on drafts, temperatures and CO_2, the human element can be accurately checked and accounted for; so there appears no good reason why the human element should control or influence the purchase of coal on an evaporative basis.

Such a chart should be an excellent thing to carry along from day to day for the benefit of the plant, even though it is not intended to buy fuel on this basis, for it will show the value of the firemen in dollars and cents.

Like all other kinds of record keeping these charts require much time to be made out. But in plants large enough to warrant checking of performances of apparatus a clerk is available who can plot curves from the tabular matter given him. Cross-section paper may be purchased that will fit nicely into large loose-leaf book covers.

Conditions Are Reversed in Making Gas—The steam engineer aims at minimum CO and maximum CO_2, while the gas producer engineer strives for maximum CO and minimum CO_2. A boiler works with a fuel bed usually varying in thickness from 3 to 12 in., whereas the depth of fuel bed in the producer varies from 2 to 10 ft. Maximum temperature in the furnace is the ambition of the fireman; on the other hand, a combustion zone of approximately 2000 deg. F., but varying with the nature of the fuel, gives the best results in the gas producer.

Philadelphia Municipal Lighting Plant—It is said that definite plans are being drawn up by the city of Philadelphia for the establishment of a municipal electric-lighting plant, which is to be ready to take over the lighting of the streets by 1916. It is understood that the plans of the Mayor, and Director of Public Works Cooke, contemplate the installation of a municipal electric plant at the old Spring Garden pumping station of the water-works, and the Keystone Telephone Co. has been asked for an option on the use of its underground conduits for the distribution system. The action was brought about by a recent decision of the State Public Service Commission, that a municipality has the right to establish a lighting system of its own without authority from the Commission, so long as it does not attempt commercial lighting.

Editorials

Engineers' Wages

Quite a few letters commenting on Mr. Pagett's article on this subject in the January fifth issue have been received, for the subject, broadly, is of interest. Nothing new or valuable is contained in these letters and for this reason we do not publish them. The writers, with few exceptions, recognize the fact that local conditions vary so widely that one should not expect to find a uniform wage over several sections of the country. These letters reflect the good judgment of power-plant men by claiming that it is right that there is no wage standard among engineers. The very nature of the service precludes such a thing if equity to all is to be had.

If an engineer's duties consisted chiefly of a few movements, physical or mental, if there were even a remote possibility of "Taylorizing" him, if there were a semblance of standardization about his routine, then well enough to talk about a standard wage. But these things cannot be. The individual's service is the only true measure of his worth.

Coal-Saving (?) Dope

There seems to be an epidemic of coal-savers on the market—not methods and apparatus for saving coal in a legitimate way, but nostrums which, sprinkled upon the coal, are claimed to greatly intensify its calorific value or at least the efficiency with which it can be burned.

As the rustic visitor to the circus said of the giraffe, "There ain't no such animal." And even as the rustic said it in the actual presence of the beast, we reiterate it in the face of claims of results produced and testimonials to savings supposed to have been accomplished.

There is no substance known to man which, sprinkled upon coal, will make it evaporate more additional water than the extra coal which the price of the dope would have bought could generate. Let us make a slight reservation. There are some coals which, thrown upon the fire, will immediately disengage a lot of volatiles like a bunch of kerosene-soaked waste. A little water sprinkled upon such coal will retard this action and perhaps save enough in volatiles which would otherwise escape, to more than offset the loss of the heat required to evaporate the water. But the action is as described and not due, as is often claimed, to the combustion of the decomposed water; for it takes just as much heat to decompose the water as it generates in getting together again.

We got caught once with one of these concoctions. We told the promoter that if he would have a test made of it by a competent and reputable engineer, and if it showed a material saving, we would publish the test and proclaim the results. He let us choose the authority, and to our astonishment the test showed from seven to sixteen per cent. better evaporation with the dope than without it.

The files of POWER will show that we carried out our promise; but even with the treated coal the evaporation was only six and a half to seven pounds, and a very little difference in manipulation would account for the bringing of a wretched performance up to this not much better one. That our skepticism was warranted is shown by the fact that the stuff was never able to hold its place upon the market.

This was many years ago. Before and since, many compounds for the same purpose have been hawked about, found a few victims and passed away. We have analyzed and exposed several of them. If they were any good, they would be in universal use now. Do not spend good money for them and be made ridiculous without something better than a salesman's claims or a lot of questionable testimonials to fall back upon when the inevitable failure comes.

Uniflow or Una-flow

The German term *Gleichstrom* (*gleich* = even, same; *Strom* = stream, current) used by German electricians for "continuous current" applies naturally to the continuous or unidirectional flow of the steam in the central-exhaust engine, reinvented and made a success by Professor Stumpf; and in German this is known as the *gleichstrom* engine. The English equivalent, unidirectional-flow, is cumbersome and soon became contracted to "uniflow." When the English translation of Professor Stumpf's book upon the engine appeared it bore the title "The Una-flow Engine." Curious as to the reason for this variation, we wrote to Professor Stumpf and to the translator.

Professor Stumpf says: "After considerable correspondence between Mr. Alexander and myself we decided upon the name 'Una-flow.' This is a little in line with, for instance, contra-flow condenser, and should be better than 'Uni-flow.' I prefer to use the hyphen, but this is a matter of taste. Uni-directional-flow engine was our first name, but we found it to be too long. Nobody would say contra-directional-flow condenser. Therefore we dropped this name and replaced it by 'Una-flow.'"

Mr. Peter S. H. Alexander, the translator of the book, says in reply: "The full term which was used by Professor Stumpf and myself was originally uni-directional-flow. When the book was fully prepared for the press in England, the English licensees, Messrs. Musgrave, had already issued a circular in which they had described it as the 'Una-flow.' In view of this, after some little discussion, it was decided, in deference to the new christening of Messrs. Musgrave, that the title of the book should be 'The Una-flow Engine.' If a short title is to be preferred to the full title of uni-directional-flow, I should say that 'Uniflow' in one word would be the best, from the point of view of everyday language. I am exceedingly sorry that there is not a more subtle or logical reason for calling the engine by the name 'Una-flow.'"

We quite agree with the translator. The English prefix for one is uni, not una. The prefix for counter or against is contra, but it is just as logical to signify the unidirectional flow of the steam in the central-ported

engine by "Una-flow," in an attempted analogy with contra-flow, as it would be to speak of the contri-flow condenser, in a forced attempt to be consistent with the other prefix.

We apprehend that Professor Stumpf knows more about inventing and designing the engine than he does about coining an English name for it, and are afraid that we cannot follow his lead in this respect, although we were inclined to adopt the spelling proposed by the man who is responsible for the success of the engine itself.

Formulas for Bumped Heads

At a recent hearing of the Massachusetts Board of Boiler Rules, it was shown that a number of the changes proposed by the board and which were considered at this hearing, were not intended as they were written. An engineer of national prominence who was present suggested that it would be well for the board to employ an engineering editor to draft such changes or additions to the rules as might be desired, so that the intent of the board would be expressed by the rules as written. The Air Tank regulations, just issued by the Board of Boiler Rules, is another evidence that the advice of this engineer was good. It is difficult to express just what is intended unless one is a master of the English language, and especially is this so when technical subjects are treated.

In the Air Tank regulations that were adopted under date of December 16, 1913, the rules were intended to be very specific as regards the calculation of the strength of bumped heads, and were drawn up as follows:

BUMPED HEADS

11. The minimum thickness of a convex head, convex to pressure, shall be determined by the following formula:

$$\frac{R \times F.S. \times P}{T.S.} = t$$

The minimum thickness of a concave head, concave to pressure, shall be determined by the following formula:

$$\frac{R \times F.S. \times P}{0.6\ (T.S.)} = t$$

R = One-half the radius to which the head is bumped;
F.S. = 5 = factor of safety;
P = Working pressure, in pounds per square inch, for which the tank is designed;
T.S. = Tensile strength, in pounds per square inch, stamped on the head by the manufacturer;
t = Thickness of head in inches.

It was unfortunate that a convexed head was referred to as one convexed to pressure, because this was contrary to the generally accepted idea on the subject; but this would not have been an insurmountable difficulty if it had not happened that the formulas were somehow reversed as applied to the two forms of heads, resulting in a higher pressure being allowed on a head which was convexed to pressure than on one concaved to pressure, as will be seen by noting the formulas given.

Soon after the publication of this set of rules, it was found that they contained a number of errors and the rules were never rigidly enforced; but the present issue was prepared after a new act of the legislature, and it was anticipated that the previous errors would be corrected. The present Air Tank rules were approved by the Board on August 12, 1914, and the subject of bumped heads was treated as follows:

BUMPED HEADS

Convex Head, Curved Outward from the Shell

12. The minimum thickness of a convex head for riveted or forged welded shells shall be

$$t = \frac{8\frac{1}{4}\ R\ P}{S}$$

except that the least thickness shall be ⅜ in. on tanks 20 in. in diameter or larger, and $\frac{5}{16}$ in. on tanks of less than 20 in. diameter.

The minimum thickness of a convex head for seamless cylinders shall be

$$t = \frac{5\ R\ P}{S}$$

except that the least thickness shall be ¼ in.

Concave Head, Curved Inward to the Shell

The minimum thickness of a concave head shall be

$$t_1 = 1.67\ t$$

where

t = Thickness, in inches, of a convex head;
P = Working pressure, in pounds per square inch, for which the tank is designed;
R = Radius, in inches = ½ the inside diameter of the outside course of the shell;
S = Tensile strength of the shell plates, in pounds per square inch;
t_1 = Thickness of a concave head, in inches.

Convex and concave heads shall be dished to a radius equal to or less than the diameter of the shell, and shall be true portions of spheres.

The description of a convex head defines what is intended and the formula given is correct as far as the evident intent to increase the safety factor on such heads is concerned; but in calculating the strength of a convex head there is no occasion to involve the tensile strength of the material of the *shell* plates of the vessel to which it is attached. That the Board of Boiler Rules believed that there was some connection between these two or that it has made the mistake of improperly expressing itself, is evident from the definition of *S*.

It will be seen, too, that an error has been made in the definition of *R*. If *R* had been stated as equal to one-half the radius in inches or, more correctly, as equal to one-half the radius to which the head was bumped, in inches, without any further additions, the formula would have been correct as far as the calculation of the strength of a bumped head was concerned. However, allowing that this error is a possible mistake of the printer, the matter is still not cleared up with the added information as given in the rule, for it will be noted that the sentence immediately below the definitions of the letters used in the formulas does not coincide with the definition of *R*. The sentence referred to provides for any radius for a bumped head which does not exceed the diameter of the shell to which the head is attached, while the definition of *R* would preclude the use of any radius which would not equal the radius of the shell to which the head was attached. It will be seen that if the rule must be literally followed as written, only a hemispherical head will be acceptable, and the value of *t* as found by the formula, will be twice as great as was really intended.

As stated in the beginning, it is difficult to write rules so that they will express just what is intended, but the employment of an experienced editor to review the rules before their publication would have avoided the errors here pointed out and would have been a real economy to the State of Massachusetts.

In every plant and factory some sort of an emergency first-aid-to-the-injured kit should be provided. A modest and yet complete one is that described on page 185, adopted as standard by the Conference Board of Safety and Sanitation. As this outfit is sold without profit, we are free to recommend it most heartily.

Correspondence

A Correction

In the issue of Jan. 5 I note in the article by Norman G. Meade, "Electromagnets for Alternating-Current Circuits," in the calculation of the magnet to work on 25 cycles, that the line voltage is taken at 440 volts. In figuring the formula for turns (T) the value 1468 turns is the number required for the entire core, or the 440 volts, and not for one coil, as stated. The computations then should be as follows:

$$Turns\ per\ spool = \frac{1468}{2} = 734$$

The ampere-turns as stated in the article are 3150 per spool. Therefore, the amperes would be $\frac{3150}{734} = 4.3$. At 2000 circ.mils per ampere this figures 8600 circ.mils. The nearest wire to this size is No. 11 B. & S., which has an area of 8234 circ.mils.

From the table given in the article, No. 11 wire has 9.7 turns per inch and, allowing 8½ in. for the length of the spool, gives 82.5 turns per layer: $\frac{734}{82.5}$ is approximately 9 layers. Assuming that the layers and the insulation between them measure 1.5 in., the length of a mean turn will be 18 in. and

$$\frac{734 \times 18}{12} = 1100\ ft.\ per\ spool$$

or 2200 ft. as the total length.

No. 11 wire has a resistance of 1.25 ohms per 1000 ft., or for the coil of 2200 ft. the resistance would be $2.2 \times 1.25 = 2.75$ ohms. Then the I^2R loss equals $4.3 \times 4.3 \times 2.75 = 50.9$ watts. The hysteresis and eddy current losses will not change and the total loss in watts will be $50.9 + 41.4 + 10 = 102.3$ watts.

W. O. Jacobi.

Omaha, Neb.

Peculiar Gas-Engine Accident

After studying over the account of the gas-engine accident as reported in the Dec. 29 issue, page 935, I cannot see how the engine could have been wrecked in any other way than by preignition or a continued too early ignition, which can readily develop from the use of a hot tube. The jacket water becoming very hot and heat radiating up around the tube guard, the flame around the tube, being better guarded, would increase in temperature. This would heat the tube to a whiter heat, which would ignite the gas at a lower compression; also, the cylinder being hot, the gas would reach a higher temperature in an earlier stage of the compression. Continued early ignition would put an unusual strain upon the housing or bedplate, and it may have been gradually fractured until one very early preignition caused it to give away.

The conditions do not indicate that the break was caused by water. In the first place, the clearance of a gas engine is nearly 20 per cent., or one-fifth the volume of the cylinder. Using illuminating gas, it is probable that the mixture was about one to eight and not lower than one to six; therefore, the gas volume of any charge would be less than the volume of the clearance. Consequently, the gas opening to the cylinder would probably not pass at any one stroke a larger volume of water than that of the gas, so that it would be impossible for enough water to pass into the cylinder in one stroke to more than fill the clearance. Moreover, if there had been enough water in the gas line to fill the opening at any one time, the gas flow previously would have been so reduced that the engine would have stopped or operated irregularly. Again, if small quantities of water had been coming over, before enough water had accumulated to interfere with the piston the cooling effect and moisture would have "killed" the hot tube, so that the engine would have continued to miss fire and stop.

The resulting condition of the engine would indicate an explosive break rather than a water break. In the latter event the strain would not have reached the breaking point until the piston was nearly in the center, in which case the breaking strain would have been in almost a straight line and the engine would not have buckled upward very much. With preignition, the break might have resulted while the piston was two-thirds or three-quarters of the way to the head center, which would leave the crank at a low angle and the strain upward. This would have the tendency to throw the shaft end forward and the cylinder upward, and the still expanding gas of the explosion would blow the piston clear out of the cylinder and upward, where it afterward fell back on top of the cylinder. If the force of the explosion had been a little greater, no doubt the piston would have been found lying on the floor in front of the engine.

L. M. Johnson.

Emsworth, Penn.

A personal inspection of this engine might disclose some peculiar reason for this accident, but if I understand the nature of the accident, one does not have to look far for the cause. It is stated that the bedplate cracked square across, and if that means a crack extending roughly in a vertical direction from a point on the frame just back of the main bearings to the bottom of the frame, the cause would seem to have been faulty design. The forces acting are exerted in a line coincident with or parallel to the engine axis, and yet, in this type of engine the metal through which the total force of each explosion reacts (the frame) is placed some distance below this axial line. The resulting action may be compared to the process of breaking a chicken's wishbone by pulling on the ends. The frame must stand a much greater stress than if the metal were placed symmetrically about the center line, and many builders do not seem to appreciate this fact sufficiently to induce them to put enough metal in the frame.

A better method is to design the frame so that the reaction is taken up by metal distributed about the line of action (the forces transmitted through the piston rod and connecting-rod); builders of large gas engines would not dare build them with any other type of frame.

Every explosion, in the type of engine illustrated, is a force tending to open up the frame in exactly the place and manner in which it evidently let go, and the intermittent application of such a force is very apt to have the observed effect in time.

The writer has seen a number of accidents of this very kind in this type. The peculiar circumstance was the position of the piston after the accident and the fact that the frame settled back into place. The rear end of the frame, together with the cylinder, usually makes a rapid rearward journey until stopped by something solid.

L. B. LENT.

Brewster, N. Y.

The break would seem to have been caused by a premature ignition. This is a common occurrence with hot-tube ignition, causing beds to break unless made extra heavy. It was only a short time ago that the writer saw a new 50-hp. engine bed break from this very cause.

G. STROM.

Titusville, Penn.

The accident was probably caused by water in the cylinder or preignition, as stated. Personally, I favor the preignition theory, as the hot tube is quite liable to vary the time of ignition—more so than the mechanically timed electrical system; or the charge may have preignited because of incandescent carbon in the cylinders, overheating, etc. It doesn't seem that enough water could have been drawn into the cylinders from the gas line to have caused any damage, as only a small part of the charge is gas.

FORREST R. CARPENTER.

Salmon Falls, N. H.

Comment on Ammonia Diagrams

I note in your Dec. 29 issue, page 930, two articles on ammonia-compressor diagrams. The first, by Charles Mugler, does not give the clearance of the compressor either in per cent. of displacement or in per cent. of the crank-end and head-end volumes. This information should be given, as otherwise it is impossible to judge whether or not the compression obtained is that for a machine in good condition. The compression curves just after the suction valves have closed seem to show an unusual increase in pressure, which may be due to piston leakage, but without knowing the clearance of the compressor it is impossible to draw correct adiabatic curves on these diagrams. If the diagrams are drawn to scale, the pressure at the end of suction for both ends of the compressor is about 19 lb., whereas for a compressor with properly designed suction valves the pressure at this point should be higher than the pressure recorded on the suction gage. This is due to the inertia of the vapor in the suction pipe which keeps the valves open even after the piston has reached the end of the stroke. The excessively high discharge pressure, even after the discharge valves are open, indicates that either the ammonia condenser to which this machine is connected is too small, or the discharge pipe too small in diameter for its length.

The fact that there is a hook at the end of the expansion line of the right-hand diagram, and not one on the left-hand one, shows that the suction valve on the right-hand end of the machine either sticks or is provided with a stronger spring than the one on the left-hand end.

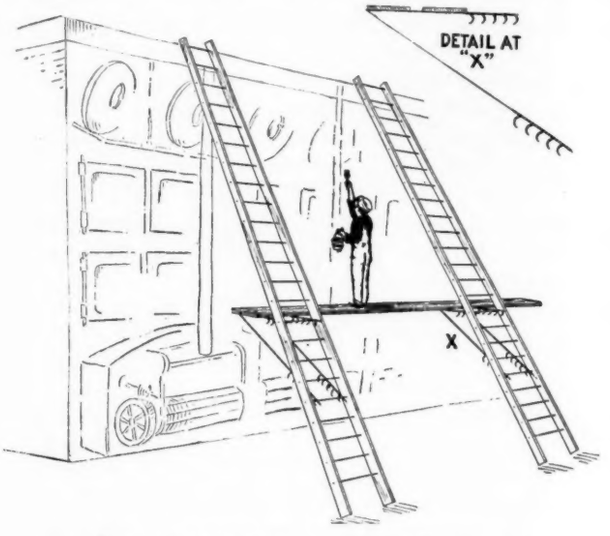
In regard to D. H. Crawford's discussion of the ammonia-compressor diagram, I do not agree with his explanation of the broken-line discharge curve *BCDE* (Dec. 29). Whenever an ammonia-compressor diagram is taken with a rather weak indicator spring, these zigzag lines are frequently noticed and are generally caused by the momentum acquired by the indicator piston from the rapid rise of pressure in the compressor near the end of the compression stroke, and seldom by the chattering of the discharge valve.

FRED OPHÜLS.

New York City.

Handy Staging

The illustration shows a handy staging for use when working on shafting, pulleys, the fronts of boilers, etc.



ONE APPLICATION OF THE STAGING

Any handy man can make and attach it to a couple of ladders without difficulty.

THOMAS SHEEHAN.

Williamstown, Mass.

Feed-Water Heaters

In the issue of Oct. 13, page 540, L. B. Carl commented on the relative merits of open and closed feed-water heaters.

The writer, having had considerable experience with feed-water heaters, begs to call attention to one statement made therein as follows: "A closed heater is not suitable, where the exhaust steam is intermittent, because the sudden changes in temperature will loosen the tubes." This is true of the straight-tube type only, and where proper provision is not made to allow for the unequal expansion of the shell and tubes.

I believe the best heater for resisting the effect of sudden temperature changes is the coil type when properly designed and built, as it will operate for any number of years without any of the joints becoming loosened, due

to expansion and contraction. All the joints should be brazed, as these are best able to stand the boiler pressure and sudden temperature changes.

W. C. Beekley.

Hartford, Conn.

The Night Engineer Off Duty

The editorial in Power, Oct. 13, 1914, entitled "The Night Engineer Off Duty" was certainly appreciated by me, for I spent three years on the night turn.

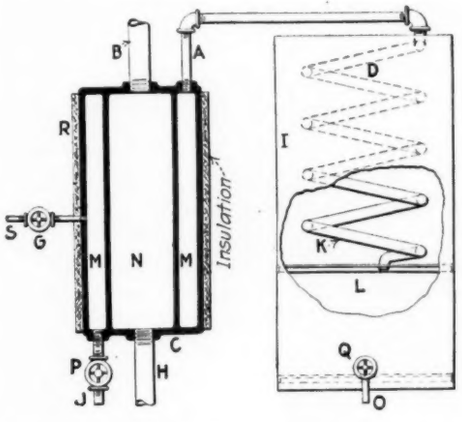
It calls to my mind a little incident of Yankee ingenuity. Several years ago, while erecting an ice plant in the South, I visited the factory one night to see how the machine was running, for we had just started and I was on the lookout for trouble. I found the old man who had the night turn sitting in a chair, holding a 12-in. monkey-wrench in his hand. As he did not seem to be using it, I asked why he was holding it, and this was the answer: It had been too hot that day to sleep much and he was sleepy. If he went to sleep he would loosen his hold on the wrench, it would fall to the floor and awaken him. He would then get up and take a look around the engine room, sit down and pick up his monkey-wrench alarm and take another rest.

Perry Losh.

Muncie, Ind.

Practical Use for Gas-Engine Exhaust

The sketch shows a water-distilling apparatus operated by the waste heat in the exhaust gases from an internal-combustion engine. It is adapted to power houses, factories and boats where pure water for drinking and other purposes is desired.



Section through Distilling Apparatus

The apparatus consists of a cast-iron drum C divided into two separate compartments N and M, a heat-insulating cover R, a cooling tank D divided into two separate compartments I and L, and a condensing coil K.

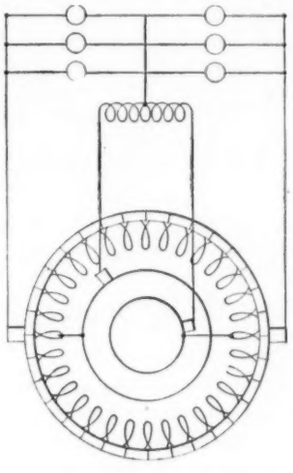
The exhaust gas from the engine enters the drum C through the pipe H, fills the chamber N, and passes out the pipe B. Water enters the compartment M, through the pipe S, regulated by the valve G, is vaporized from the heat in compartment N, passes through the pipe A into the condensing coil K, surrounded by cold water in the tank I, is condensed, and flows into the reservoir L, whence it is obtained from the faucet Q. The residue from the distilled water is run off through the pipe J, controlled by the valve P.

F. B. Hayes.

Houston, Tex.

Flickering Lights

The sketch shows the connections of a three-wire generator. For simplicity it is shown with two poles, although four or more are usual. The armature generates 220 volts, obtainable from the outside wires connected to the brushes. Two slip-rings mounted upon the armature shaft are connected at diametrically opposite points of the armature winding, and from brushes bearing upon the slip-rings conductors are carried to the ends of an iron-cored reactance coil, to the center of which is connected the middle or neutral wire of the three-wire system. A resistance might be used for this purpose, but as the device is continuously subjected to alternating e.m.f., reactance is more effective in limiting its value.



Connections of Three-Wire Generator

The net result of the arrangement is that 220-volt motors may be operated from the outside wires and 110-volt lamps from either outside wire and the neutral. The reactance carries direct current only when the two sides of the service are unbalanced. The unbalancing direct current entering the reactance at the center divides, half flowing around the core in one direction and half in the other; its magnetizing effect is, therefore, practically nothing.

An inspector was called to find out why the lamps fed by such a unit flickered. Inspection of the taps from the rings to the winding disclosed that they were not tapped to the winding at equidistant points. Changing the taps to points of symmetry stopped all flickering.

J. A. Horton.

Schenectady, N. Y.

Drains above Back-Pressure Valves

A drain should be connected just above the back-pressure valve. If connected at a higher point, the vapor may condense and create a static head above the valve, which will prevent it opening under ordinary pressure.

The vent pipe may be dispensed with by drilling small holes in the seat of the back-pressure valve. The escape through these openings will be sufficient to relieve air-binding in the heater and drain back any condensation in the exhaust or vapor pipe.

T. W. Reynolds.

New York City.

Unsafe Blowoff Piping

A couple of years ago I was a fireman and assistant in a small cold-storage plant. There were three boilers, but we only fired one, which was sufficient to carry the load. The night fireman asked me to do all the blowing down on No. 2 boiler, as its blowoff valve was situated so far back in a dark corner that if anything should happen he wouldn't have as good a chance to get out in the dark as I would in daylight.

I did so until one day the boiler inspector told us to disconnect No. 2 blowoff, as he did not believe the pipe between the valve and boiler was made up very tight. You may imagine our surprise to see it drop out of the elbow near the boiler after giving it only a half turn when unscrewing it. The pipe had only two threads caught, and they were nearly eaten out. The steam fitters had cut it too short, but used it anyway to save cutting another. I learned afterward that the night fireman knew of this, hence his distaste for blowing down this boiler.

C. KNOWLAND.

Louisville, Ky.

Why the Gage Hand Vibrated

The cause of the gage hand vibrating, as referred to by A. E. Aldrich in the Jan. 5 issue, was the intermittent steam flow caused by the cutoff of the reciprocating engines. During the daily periods referred to there was some change in conditions, as an additional unit in service or *vice versa*. The oiler in turning the valve simply closed it a little more than usual, which should have been done before.

JOHN F. HURST.

Louisville, Ky.

Cost of Operating Vacuum Ash-Handling Systems

The discussions of vacuum ash-removal systems in the July 7, Sept. 8 and 15, and Oct. 20 issues, following the article describing the Girtanner-Daviess system in the April 7 number, have been both interesting and fair-minded and have brought out a number of instructive features. The point is to be emphasized, however, that the instances of expensive installation and heavy repairs cited do not refer to the system described in the original article, as the discussions refer to motor and blower costs instead of a steam jet. Even Mr. Sandstrom's estimate of charges in the July 7 number is based on his experience with a blower system.

The absolute cost per ton of ashes removed has no comparative value as a criterion. The most economical system, from wheelbarrow up, for an unfavorable location, may still leave costs high. Furthermore, the final receiving tank may be ignored for purely comparative costs of different systems (unless special expense is here necessitated by the peculiarities of a system) since such tank should rather be imposed equally on each system by the final disposition made of the ashes. This system will discharge directly onto a dump or into any receptacle.

The maximum repair bill for a year on steam-jet installations ranging in price, for pipe line only, from a few hundred up to $1400 has been a fraction over $32—about one-seventeenth of the 40 per cent. experience of Mr. Sandstrom. Again, one man is well able to handle the seven tons per hour for which Mr. Sandstrom wishes to hire two, and this cuts his labor cost in half. In moderate-sized plants the system may be operated by the fireman along with his other duties, and in larger plants the attendance is a minimum. The initial cost is low, so that interest on the investment does not eat up economies secured. The steam used in the jet amounts to 5.29c. per ton (see Apr. 7 POWER).

GIRTANNER-DAVIESS.

St. Louis, Mo.

Finding the Best Coal

The letter, "Finding the Value of Coal," by William A. Dunkley, in the Jan. 5 issue, interested me. One point of importance was not stated in the article, and that is, whether the coals furnished by the roads A and B were of sensibly the same character. If the fuel tests of the two coals showed wide variations in ash, volatile, sulphur and heat, trouble in the fire-room could have been predicted in advance of the change from one coal to the other, provided no change was to be made in the method of handling the fires.

The following quotation from a paper read by me before the New Jersey Clay Workers' Association at the winter meeting at New Brunswick, N. J., Dec. 29, 1914, should serve to make clearer the fundamentals involved in the problem which confronted Mr. Dunkley:

> . . . The combustion of coal in a furnace is a complex process, and the different combinations of equipment and methods of handling the equipment are almost infinite. And there are numberless kinds of coal. That statement is made advisedly. Considering alone the inherent characteristics, that is, the sulphur, volatile, ash, the fusing point of the ash, and the heating value of the coal, you can, within certain limits, find coals with all of these factors in any proportion you may desire. Here are five variables and each one has a considerable range of variation. In addition there is the variation in the physical condition of coal; that is, whether it comes in the form of dust—the slack—or screened to a certain size, or what is called the "run of mine." Then there is the question of coking or noncoking.
>
> There are few, if any, industrial uses for coal which require definite adjustment of every one of these variables. Certain definite limits of part of the characteristics already mentioned are especially required in any particular case, without, however, limiting the remaining factors. The first point is, therefore, that you must know what factors are material in the selection of your coal before you can change to a new coal with any reasonable expectation of getting better results, or the same results at less cost.
>
> The chief use of coal is to produce heat by its combustion. In any particular process to accomplish the best results a certain amount of heat must be released in a certain length of time. Let us assume for the present that this is the whole problem, disregarding all other questions, such as the avoidance of smoke, fumes injurious to the product, etc. That limits us to three variables—the heating value of the fuel per pound, the amount of fuel burning at one time, and the amount of fuel burned per hour. The first is determined by the selection of the fuel, the second by the area of the grates, and the third by the thickness of the fire and the amount of air supplied to it. It is obvious that if we change any one of the three, we will directly affect the result, which is a given amount of heat liberated in a given time. This may seem rather elemental, but the point I wish to bring out is that frequently the user tries to change one of these factors (usually the coal) without making corresponding readjustments in one or both the remaining conditions, with the result that he concludes there is only one kind of coal he can burn satisfactorily. Frequently, a minor change in equipment or methods of firing will make possible the use of another coal at considerably less expense.

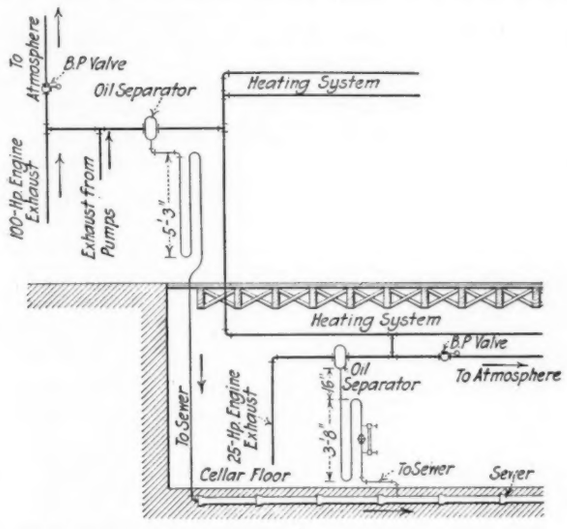
CARLTON W. HUBBARD.

Brooklyn, N. Y.

Trouble with Oil Separator

Can any POWER reader suggest a remedy for trouble with an oil separator? Since taking over my present plant I have had considerable annoyance from grease passing over with the steam into the heating system from the separator of a small engine located in the cellar, where the connections are arranged as in the accompanying sketch.

A new separator and various kinds and sizes of seals



ARRANGEMENT OF OIL SEPARATOR WHICH GIVES TROUBLE

have been tried without effecting a remedy. We carry 8 in. of vacuum and the grease is drawn through the separator. Several so called experts have seen the conditions and each has advised different sizes, loops, etc., but upon trial none of them has proved satisfactory.

H. G. GOODWIN.

Lachine, Que.

Industrial Education Again

My article on industrial education for operating engineers, which was published in POWER, July 28, 1914, was intended to draw out discussion. I was glad, therefore, to hear from William E. Dixon, in the Nov. 10 issue, even though he does not agree with me and thinks the sort of trade school I was connected with so long is a "pure and arrant fake."

This problem of industrial education is far from being solved; those who have worked the hardest and the longest realize that better than those who are on the outside looking in. Many mistakes have been made, and many more will be made. The fact remains, however, that the industries are today so organized that learning any trade under the old conditions is practically impossible. The few large shops which realize this fact are providing schools within their own walls. These have all the problems to meet which are met by the public trade school, if politics can be kept out of the latter, and that is possible.

There is no need to forget the cultural, civic or business side of a boy's education because he is in a trade school. I do not know what schools Mr. Dixon has seen, but his name is not on the visitors' book at the Worcester Trade School, so I am sure he has not been shown that school by anyone in authority able to explain what is really done there. While some things are far from perfect, they are not those to which he alludes. These schools should, and most of the state-aided schools do, give at least one-third of the pupils' time to regular high-school studies. Their hours are much longer, and consequently this time is almost equal to the total time which the regular high schools give.

Some interested in industrial education do believe a trade school should teach trades solely. We have never felt that we could conscientiously send out a graduate unequipped, so far as his mentality allows, for the battle of life.

It is not easy to find the right teachers. We had one for a short time who had a first-class license. He had handled some big plants and came well recommended. He knew nothing except by rule of thumb, could not tell that to anyone else, and would not be helped by those who did know how to teach.

A man who knows what he is trying to teach, who has not forgotten that he was a boy, and who has a cheerful disposition has a fair chance of learning to teach by experience alone. Certain tricks of the trade of teaching he may learn from more experienced men. These relate almost entirely to discipine. The largest job is how to impart what one knows to students so that it makes an impression and is within their comprehension. When an engineer has done a certain thing for years he easily loses sight of why he does it. It is enough to tell a helper to do a thing, and he does it until it becomes a habit. That is not education. A man has not learned how to do a thing until he understands the reasons for doing it.

Mr. Dixon should prove his statement that "the schools cannot even hope to get an equipment that would serve for much more than a toy." There is no reason why a school's money should not go as far as that of any corporation. In fact, experience shows that it can be made to go farther than any other city money. Builders of the best machinery have made concessions that, taken alone, would fit out a school in far from toy fashion. The cost per pupil for adequate equipment is less than that for equally good machine-shop equipment.

Mr. Dixon hints that desirable boys will not go to school to learn to be engineers. That depends on what you will take. If you start a school with riffraff it will be hard to get decent boys into it. A school cannot afford to be snobbish, but it can make a manly attitude toward the work a condition of membership. There are plenty such boys, but they will not go where the other kind are tolerated in any considerable numbers.

"It is bad practice to try to teach a person an idea until that person has a desire to know it," says Mr. Dixon. If my father had humored me when I was a small boy, I would not have begun to get ideas yet. A boy should be taught while he is young and receptive. The instructor is of little use who cannot make steam engineering attractive to a boy. A boy must be brought to the point where he knows that it is up to him to learn before he will learn many things.

The crying ill of all our educational systems is that the pupils do not wake up to their needs until it is too late. They are so delicately handled now that they think they know much better what is good for them than their parents. The adequate substitute for the stiff stick in the

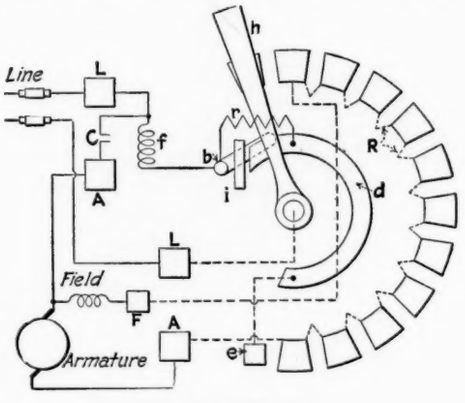
hands of a raw-boned Yankee teacher as an idea awakener has never been found. If this notion that a boy should not be taught anything until he has discovered for himself that he needs to know it were ever followed except on paper, our schools would be picnics and education would be below zero.

E. H. FISH.

Worcester, Mass.

Contactor Closed and Opened

The diagram shows the connections of a starting-box used in conjunction with a contactor which ruptures the arc, thereby saving the rheostat contacts when the starter is thrown to the off position. It also acts as a no-voltage release to open the motor circuit should the line become dead, with no one in attendance. Upon simul-



STARTING DEVICE

taneously twisting handle *h* and moving it in a clockwise direction, an auxiliary contact arm mounted on *h*, and brought down by the twisting motion, touches the button *b*, to which is connected one end of the operating coil *f* of contactor *c*. This closes the local break through the contactor and the motor takes current. If the handle is kept twisted and in contact with *d*, as it is advanced, coil *f* will remain energized throughout the travel of the handle. At the end of its travel the handle engages a spring clip on post *e* and the twisting stress may be released, because coil *f* is then energized independently of the auxiliary contact.

It will be noted that at the start, when the auxiliary contact rests upon button *b*, the full line voltage is applied to the coil *f*, but after the contact reaches segment *d*, resistance *r* is cut into series with coil *f*. This is a feature of safety as well as of economy, because less current is required to hold the contactor closed than is required to close it; and if the voltage should leave the line, thereby causing the contactor to open, it would remain open until someone returned the starter to the "off" position and repeated the starting cycle, because with *r* in series with *f*, the current would be insufficient to close the contactor.

One of these outfits in a lumber camp, where it was exposed to the weather, was complained of because in going from the "off" to the "on" position the contactor would open immediately after closing. Insulation strip *i* is used simply to preserve a smooth surface for the auxiliary contact in its travel from button *b* to the segment *d*. Investigation showed the trouble to be due to this insulating strip having absorbed rain and so swelled as to raise the auxiliary contact finger out of contact with *b* before it had made contact with *d*. This, of course, demagnetized coil *f* and permitted the contactor to open. The trouble was remedied by sandpapering the surface of *i* below the surfaces of button *b* and of segment *d*. It is interesting to note that an open-circuit in resistance *r* may cause the same failure.

J. A. HORTON.

Schenectady, N. Y.

Loss by the Use of Slack Coal

Here is an account of two boiler tests that may interest some engineer operating the Hawley down-draft furnace. Plants with these furnaces have their troubles when using slack coal. When we have mixed coal we can show good results, but with slack we cannot keep it on the top grate. The result is that we cannot carry the load with the same number of boilers on the line. With mixed coal I ran a test on Dec. 2, 1914, on a 150-hp. Coatsville fire-tube boiler with a Hawley down-draft furnace. The first test was very good, the evaporation from and at 212 deg. F. per pound of dry combustible being 11.66 lb., the horsepower-output 287, an over-rate of 91 per cent., and the efficiency of the boiler and furnace was 69 per cent. It cost 10.21 cents to evaporate 1000 lb. of water.

The following day I ran a test on the same boiler, with the same fireman, and tried to get as near to the first test as was possible, but using coal containing very few lumps. This second test showed a decided falling off. We evaporated 10.63 lb. of water per pound of combustible and developed 249 hp., an over-rate of 66 per cent. in place of 91 per cent. It cost 11.25 cents per thousand pounds of water in place of 10.21 cents and the efficiency was only 64 per cent. The loss in twenty-four hours on nine boilers amounted to $21.33 with slack coal.

HARRY BIEHL.

Philadelphia, Penn.

Commutator Short-Circuited

An inspector was called to find out why the armature of a motor was heating and "shooting like a gun." He ascertained that the commutator had just been slotted in a lathe. The side mica was very thin and the lathe tool used was too thick; the result being that the tool curled shavings from the bars and at the end of the stroke the stub of the shaving was jammed into the mica and across an adjacent bar. The burrs had been picked out where they could be seen, but some were too deeply embedded.

As soon as the motor was started, all the coils that were short-circuited by bridged bars were heated by the local short-circuit currents which, as the speed became greater, became sufficiently heavy to burn out the short-circuits that had not been picked out. The noise like the report of a gun was due to the extinction of the arc by the magnetic field. The motor was then run up to full speed and without any further demonstration. The arc-extinguishing properties of the magnetic field explain why armatures that burn out in service are not as badly damaged as might be expected.

J. A. HORTON.

Schenectady, N. Y.

Inquiries of General Interest

Salt in Fireclay—What is the purpose of adding salt to fireclay used in setting the firebrick of boiler furnaces?

M. G. W.

The addition of about a pint of salt to a bucketful of ordinary fireclay causes partial vitrification of the fireclay when it becomes heated, and increases its adhesiveness to the firebrick.

Removing Stains from Gage-Glasses—How can stains be removed from boiler water gage-glasses?

S. M. E.

Most stains formed on boiler gage-glasses can be removed with a swab of clean waste moistened with a weak solution of muriatic acid. In the cleaning process care should be taken, however, not to employ wire or other material likely to scratch the glass and thereby weaken it.

Coefficients of Expansion—What are the relative rates of expansion of aluminum, brass and cast iron for the same increase in their temperature?

C. R. H.

The coefficients of expansion or proportionate increase of length for each degree increase of temperature of the metals named are: Aluminum, 0.00001234; brass, 0.00001; and cast iron, 0.00000556.

Bedding Underground Pipes in Sand or Gravel—What benefit is to be derived from bedding underground steam or water pipes in sand or gravel?

J. N. R.

The principal advantages obtained are more perfect grading and better subdrainage. In case of steam pipes, whether or not they are laid in conduits or coverings of any kind, subdrainage is important in reducing the convection of heat from the pipes to the surrounding earth, and in case of water pipes complete bedding in sand or gravel affords better protection against frost.

Drainage of Boiler Steam Main and Arrangement of Stop Valve—How should a main steam pipe slope and boiler stop valve be placed between a boiler and engine?

D. J.

The slope of the piping and arrangement of the stop valve should be such that all condensation between the valve and the boiler will drain back into the boiler, and the slope of the piping beyond the valve should be such that the water will drain from the boiler toward the engine. Stop valves should be so arranged that pressure from the boiler will tend to raise the valve from its seat, and they should be given such a position that there will be no accumulation of condensate above the valve when it is closed.

Care of Standing Boiler—When a boiler of a battery is not required for some time, is it injured by standing with water at the usual level carried for steaming?

W. B.

There will usually be more rapid corrosion of the interior above the regular water line, and especially near the water surface, when so standing than when steaming. When not required for some time, the boiler is better preserved by emptying it of water and drying thoroughly. If that is impracticable, interior corrosion can be reduced by completely filling the boiler with water, although for most situations this results in more rapid exterior corrosion than when the boiler stands empty, due to condensation of the moisture of the atmosphere.

Pressure in Discharge Pipe with Drop Leg—Neglecting pipe friction and inertia, what pressure would a pump have to work against if the discharge pipe rose to a height of 150 ft. and returned with an open end at the level of the pump?

G. K.

For starting flow in the descending leg, the pressure pumped against would increase to

$$150 \times 0.434 = 65.1 \text{ lb. per sq.in.}$$

As a column of water 34 ft. high would balance the pressure of the atmosphere, then if a solid column 34 ft. or more in height were maintained in the descending leg of the discharge pipe, a vacuum would be formed in the upper end of the ascending pipe, thus relieving it of pressure equal to that of the atmosphere. The net head pumped against would then be 150 — 34 or 116 ft., which would be equivalent to

$$116 \times 0.434 = 50.34 \text{ lb. per sq.in.}$$

Coal Required under Stated Conditions—With the temperature of feed water at 200 deg. F. and a combined boiler and furnace efficiency of 60 per cent., how many pounds of coal of a calorific value of 13,500 B.t.u. per lb. would be required to evaporate 13,600 lb. of water into dry saturated steam at 130 lb. gage pressure?

T. B.

The total heat required to convert a pound of feed water from 32 deg. F. into dry saturated steam at 130 lb. gage or 145 lb. absolute pressure would be 1192.8 B.t.u., and as each pound of feed water at 200 deg. F. would contain 200 — 32 or 168 B.t.u. above 32 deg. F., then to raise 13,600 lb. of water from 200 deg. F. to dry saturated steam at 130 lb. gage pressure would require

$$13{,}600 \times (1192.8 - 168) = 13{,}937{,}280 \text{ B.t.u.}$$

With a combined boiler and furnace efficiency of 60 per cent., from each pound of coal containing 13,500 B.t.u. there would be realized

$$13{,}500 \times 0.60 = 8100 \text{ B.t.u.}$$

and, consequently, the evaporation of 13,600 lb. of water under the conditions stated would require

$$13{,}937{,}280 \text{ B.t.u.} \div 8100 \text{ B.t.u.} = 1720 \text{ lb. of coal.}$$

Pipe Surface Required for Heating Water—How many lineal feet of 2-in. iron pipe, or of brass or copper pipe of the same size, would be required as heating surface in a closed tank to heat 800 gal. of water per hour from 50 to 180 deg. F. with exhaust steam at 1 lb. gage pressure?

A. M. D.

In raising the water from 50 to 180 deg. F., each pound would receive 180 — 50 or 130 B.t.u., and as 800 gal. of water would weigh $800 \times 8\frac{1}{3} = 6666$ lb., and, neglecting losses by radiation, the total heat to be transferred would be

$$6666 \times 130 = 866{,}580 \text{ B.t.u. per hr.}$$

As the temperature of the steam would be about 213 deg. F. and the average temperature of the water would be

$$\frac{50 + 180}{2} = 115 \text{ deg. F.}$$

then the mean temperature difference between the steam and the water would be

$$213 - 115 = 98 \text{ deg. F.}$$

For this mean temperature difference, iron pipe would condense about 18.5 lb. of steam per square foot per hour, and the latent heat of steam at 1 lb. gage pressure being 969.7 B.t.u. per lb., then for each square foot of pipe surface there would be a liberation of

$$18.5 \times 969.7 = 17{,}939.45 \text{ B.t.u. per sq.ft.}$$

so that

$$\frac{866{,}580}{17{,}939.45} = 48.3 \text{ sq.ft.}$$

of iron-pipe surface would be required. As 1.608 lin.ft. of 2-in. pipe would be required per square foot of external surface, the total heating surface would require

$$1.608 \times 48.3 = 77.7 \text{ lin.ft.}$$

of 2-in. iron pipe. Under the same conditions brass pipe would condense about twice as much and copper pipe about $2\frac{1}{3}$ times as much steam per square foot of surface, and therefore

$$\frac{77.7}{2} = \text{about } 38.8 \text{ lin. ft.}$$

of brass pipe, or

$$\frac{77.7}{2.33} = \text{about } 33.3 \text{ lin.ft.}$$

of copper pipe, of the same external diameter as standard 2-in. iron pipe, would be required.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—EDITOR.]

Engineers' Study Course

Steam-Engine Cycles

The diagram representing the ideal performance of the steam plant is given in Fig. 1, repeated from Fig. 3 of the article on "Heat-Engine Cycles." A brief review of this Rankine cycle is as follows:

Line *AB* shows complete evaporation and the transfer of the steam to the engine or turbine, without loss of heat by radiation, loss of pressure by pipe friction or throttling, or loss of volume by initial condensation. Its right-hand end may also cover superheating, which, of course, takes place at boiler pressure.

Curve *BC* shows adiabatic expansion, possible only in a cylinder of some imaginary, thermally neutral substance. This expansion is carried clear down to the exhaust pressure at *C*.

Line *CD* represents complete expulsion of the steam from the engine and its contraction to the liquid state in the condenser (or atmosphere).

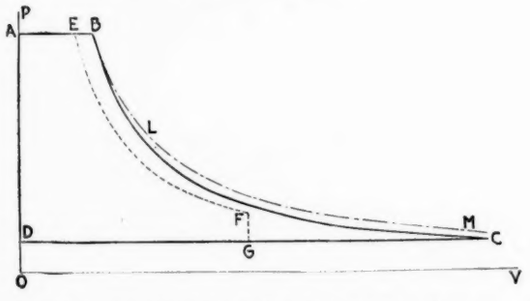


FIG. 1. DIAGRAM REPRESENTING IDEAL PERFORMANCE OF STEAM PLANT

This diagram implies that the engine has no clearance at all. It is the form of ideal action in either piston engine or turbine, but the interpretation is somewhat different for the two types of machines. The engine will now be considered.

Area *ABCDA*, Fig. 1, shows the maximum output of work per pound of steam, within the particular limits of pressure and temperature, and the best possible efficiency. In the actual plant there are four ways or directions in which this ideal performance fails of realization. These sources of loss are:

(1) Pipe and valve losses, of heat and pressure, incurred in the transfer of steam to and from the cylinder.

(2) Thermal action of the cylinder walls.

(3) Incomplete expansion.

(4) Clearance and compression.

Of these, Nos. 2 and 3 will first be taken up, then No. 4, and finally No. 1; and in the consideration of them the evolution of the actual indicator diagram from the ideal outline *ABCDA* will be shown.

The effect of initial condensation, by the cooler metal surfaces with which the steam entering the cylinder comes into contact, is evidenced in the shrinkage of steam volume from *AB* to *AE*. And then, because the heat thus taken from the steam at high pressure begins to come back as pressure and steam temperature are lowered, curve *EF* falls less rapidly than would an adiabatic from *E*. In shape, curve *EF* is here drawn as an equilateral hyperbola, following the law

Pressure × *volume* = *a constant*, or $pv = c$.

Sometimes the hyperbola is called the theoretical curve of steam expansion. The title is undeserved, for thermal conditions within the cylinder are so complex that the formulation of any theory of expansion is impossible. The very prevalent use of this curve in laying out preliminary or illustrative diagrams is based on two wholly practical facts or considerations. The first is that the hyperbola is a fairly good working average of the expansion curves of actual indicator diagrams; the second, that it is an easy curve to plot.

The best collection and discussion of data as to the form of real steam curves that have been made will be found in the paper on "Cylinder Performance," presented to

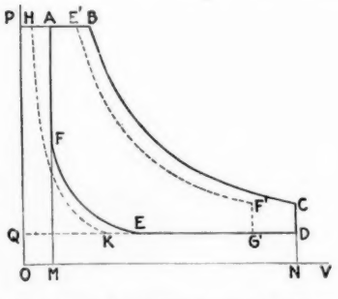


FIG. 2. ILLUSTRATING CLEARANCE AND COMPRESSION

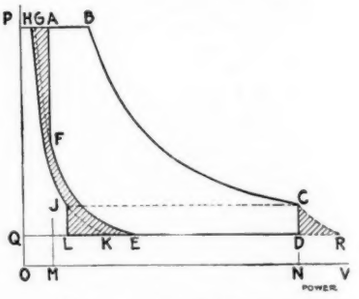


FIG. 3. LOSS DUE TO CLEARANCE

the American Society of Mechanical Engineers by J. Paul Clayton, in May, 1912, and reviewed in POWER for June 18, 1912. The subject is too extensive for more than a reference here. It is enough to say that when there are conditions favoring excessive cylinder-wall action, such as small size and low speed, with early cutoff, the expansion curve will run much above the hyperbola. On the other hand, with high superheat and small thermal action, it will fall much more rapidly. But in the general run of ordinary conditions, departures from the form $pv = c$ are comparatively small.

Returning to Fig. 1, it will be noticed that the assumed curve *EF* rises steadily toward the adiabatic *BC* as the steam expands. In further illustration of the same point, curve *LM* is a hyperbola drawn from *B*. The vertical distance between *LM* and *BC* at first grows larger, then diminishes; but when it is remembered that this difference is a relative quantity, to be compared with the whole pressure from base line *OV* up to the curves, it is seen to increase progressively.

Now in ideal operation, or in the process reasoned out for getting the maximum work from a pound of steam, expansion is carried clear down to exhaust pressure. There is very good reason why this ought not to be done in the real engine, and why it is more economical to stop

at some such point as *F*. To get the small amount of work represented by the triangular area to the right of line *FG* would require a cylinder more than twice as large as is needed to contain volume *DG*. First of all, this would make the engine cost more, but worse than that, it would involve continual losses in operation. If the cylinder is too big and cutoff too early, the waste due to thermal action by the cylinder walls becomes relatively greater. And in driving the piston by a small mean effective pressure such as will prevail beyond *FG*, the loss of work through machine friction will exceed the effective work done by the steam upon the piston, and the result will be a net loss rather than a gain.

The matter of clearance and compression is taken up in Fig. 2. As the first step, the combined expansion and release lines *E'F'G'* are transferred directly from Fig. 1. These lines represent the performance, in an engine without clearance, of one pound of steam which enters the cylinder, does work, and goes to the exhaust. But now there is to be associated with this working steam a certain proportionate amount of clearance steam. In Fig. 2, the volume *PE'* of the working steam in its condition at cutoff is moved out to *HB*, leaving back of it a space *PH*

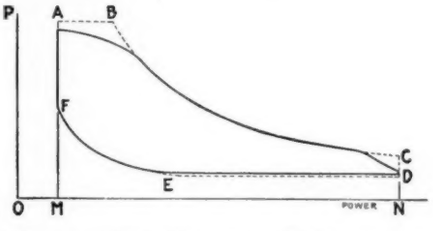


FIG. 4. THE DIAGRAM OBTAINED IN PRACTICE

filled with clearance steam. Of course, this division is imaginary, for there is no separation into distinct volumes. The point is that, of the steam present at cutoff, a portion *HB* is going to be discharged, while the remainder *PH* will be caught and compressed.

As the whole body of steam expands along curve *BC*, the clearance quantity has its increasing volume measured out to the similar curve *HK*. Since these curves are taken to be of the same form as *E'F'*, horizontal distances between curves *HK* and *BC* are the same as between line *OP* and curve *E'F'*. In effect, then, the original no-clearance diagram *PE'F'G'Q* is shifted over to the right of curve *HK*. This distorts its shape, but does not change its area above a horizontal line through *F'* or *C;* below that pressure, however, there is a loss, the cause of which can be stated in two ways. The first is, that *F'G'* would be changed to a curve *CR*, as shown in Fig. 3, at a constant distance from curve *HK;* and the vertical release line *CD* cuts off the extended area *CRD*. The other form of statement is, that since effective volumes of the working steam are measured over from curve *HK*, and these grow shorter below the terminal pressure at *C*, the lower end of *HK* cuts under the effective diagram and diminishes its area.

The loss due to clearance is more fully illustrated in Fig. 3. At the beginning of expansion the steam in the cylinder is partly condensed, because of wall action. At the beginning of compression the steam left in the cylinder is likely to be nearly or quite dry, perhaps even a little superheated. Consequently, the quality of the clearance steam is higher at *E* than at *K*, and its volume is greater; then the compression curve runs to the right of *HK*, or the clearance steam requires more work for its compression than it gives back in expansion.

The lost area *CRDC* is, in effect, an addition which compression makes to the work not utilized because of incomplete expansion. This triangular figure may be carried over to the position *JKLJ*: and then we say that while the clearance steam really follows curve *HK* all the way down to exhaust pressure, its effective delivery of work ends at the pressure of release. Any work of expansion to the right of *JL* simply helps the outrush of steam during release. A condensed statement regarding clearance losses may be made as follows:

If the whole weight of steam represented by volume *PB* came from the boiler into an engine without clearance, it would do the work represented by area *PBCDQP*, Fig. 3.

Actually, because of clearance and compression, the useful work really performed is only that represented by area *ABCDEFA*.

But the area *PAFEQP* is not all loss, for a part of it is covered by the work *PHJLQP* of steam which did not come from the boiler, but was saved over from the preceding cycle. The net loss is then the shaded area *AFELJHA*.

This lost area is made up of three parts. In order to separate them, the compression curve is extended as *FG*, here made similar to *HK*. This continues up to admission pressure, the prevailing difference in quality between compression and expansion, and shows a complete working cycle *EGHKE* for the clearance steam. Then the three partial losses are:

Area *AFGA* due to throttling of the live steam as it enters and fills the clearance space.

Area *EGHKE* due to cylinder-wall action, working out through the cycle of operations of the clearance steam.

Area *JKLJ*, in effect, as has been explained, an addition to the incompleteness of expansion.

In regard to the proportions of this diagram, it is to be noticed that the difference between curves *EG* and *HK* is exaggerated, being too great relative to the quality at cutoff shown by the ratio of *AE* to *AB* in Fig. 1.

The ideas developed in Fig. 3 open the way to a more or less definite rational determination of the best degree of compression corresponding to a certain set of conditions on the expansion side of the indicator diagram. The indefiniteness is due in part to uncertainty as to the exact form of the expansion and compression curves in the actual engine, and for the rest to modification of the sharp-cornered diagram by the pipe and valve effects shown in Fig. 4. But this general idea of the several sources of loss makes it easier to understand the results of experiments made to determine the effect of compression upon economy.

In Fig. 4, the outline *ABCDEFA* is the same as in Figs. 2 and 3, but its proportions are changed to something nearer those of common indicator diagrams. It only remains, then, to sketch in curves of admission and release and a line of increased back pressure, and thus come to the end of the evolution of the actual indicator diagram. Knowledge of the magnitude of these pipe and valve effects comes wholly from experience, or from familiarity with the performance of the various classes of engine.

Performance of Refrigeration Plant at Lubeck, Germany*

SYNOPSIS—Description and data of test results of the municipal refrigeration plant at Lubeck, Germany. The poppet-valve compressor engines use highly superheated steam and some exhaust steam is used for ice making. Unusually good results are obtained.

In contracts for ice-making and refrigerating machinery it is customary for the manufacturer to guarantee the capacity, to enumerate the temperatures to be maintained in cold-storage rooms, and, if the purchaser is exacting, to insert the guarantees for coal, steam, power and cooling-water consumption. A test is usually made to see if the machinery supplied fulfills the various requirements. Such tests are instructive, but for commercial reasons, probably, they seldom get into print.

In the following only the acceptance tests of the equipment of the refrigeration plant for the city of Lubeck, Germany, are given.

The mechanical equipment of the "Kuhlhaus Lubeck" consists essentially of one 17¾ and 29½ by 29½-in. tandem-compound Swiderski steam engine of 320 hp., using superheated steam, connected to a Balcke surface condenser. In the connection between the low-pressure cylinder and the surface condenser is inserted an oil separator. From the condenser the condensate is taken to a reboiler for the purpose of expelling air and foreign gases, a portion of this water being required for making distilled water ice, and the remainder is returned to the boiler. The distilled water is forecooled in a countercurrent cooler.

This main engine is coupled to a pair of 13x23⅝-in. double-acting horizontal Borsig ammonia compressors of the opposed type. A duplicate engine, intended for reserve, is connected to a single ammonia compressor 13x23⅝ in.

To take advantage of the increased capacity obtained when operating with a higher evaporating or suction pressure, one of the four ends of the double compressor unit is used only for ice making, keeping the brine in the freezing tank at about 19 deg. F. The guaranteed capacity of this compressor cylinder half is 36.4 tons or

110,000 calories × 3.9683 = 436,513 B.t.u. per hour.

The ice-tank room and brine tank are designed on the Linde system, the brine-cooling coils being placed underneath instead of between the cans. The rated daily capacity is 22 tons of 2000 lb. each.

With the 300- or 400-lb. American blocks the freezing time is 42 to 60 hours, while the small European 55-lb. blocks (average size 7x7x35 in.) with 19-deg. brine freeze in 18.4 hours; there are 612 of these small cans in the freezing tanks.

When cooling brine to 14 deg. F. for the cold-storage warehouse the guaranteed capacity of any one of the compressor cylinders is 61.15 tons of refrigeration. (One ton of refrigeration = 2000 × 144 B.t.u. = 288,000 B.t.u. per 24 hours.) The brine is cooled by direct-expansion coils. The brine tank has circulation partitions and agitators. Another tank, supplied with cold water, is used for cooling the liquid ammonia. The ammonia condensers of the atmospheric type are on the roof.

From the flywheel of each engine a belt leads to a lineshaft under the engine-room floor, and from this lineshaft are driven two 70-kw. Siemens-Schuckert generators, one being a spare. The boiler house has two Borsig water-tube boilers, one a spare, built for 170-lb. pressure and superheating to 625 deg. F. Each boiler has 969 sq.ft. of heating surface.

The cooling water supply, guaranteed capacity 268 gal. per minute, is obtained from two wells. The water in one is lifted 131 ft. to the surface, in the other 98 ft., by a Borsig air compressor, also in duplicate. The water is discharged into a receiving basin near the boiler house, and forced to the ammonia condensers by a belt-driven volute pump. A duplicate pump, driven electrically, is provided for this service. As a further safeguard for insuring uninterrupted operation, provisions are made for using city water if necessary.

For producing the necessary hot water for scalding purposes in the abattoir a steam-heated hot-water apparatus is erected in the boiler house; capacity 26.5 gal. of water per minute heated from 86 deg. to 176 deg. F. Preheated water is taken from the surface condenser, the latter in turn receiving its cooling water from the ammonia condensers. In this manner the water is utilized to the fullest extent. Steam meters measure the quantity of live steam fed to the hot-water apparatus. The plant is well equipped with all kinds of indicating and recording instruments.

The following is abstracted from the report of acceptance tests made after all the machinery had been installed and operated during the summer of 1913.

BOILER TEST

Of the two boilers, which are alike, only the one which happened to be clean at the time was tested, it being assumed that the other boiler would have shown equal efficiency.

Date of test	Oct. 18,1913
Type of boiler (Borsig Steilrohr Kessel)	Water-tube
Number of boiler and year built	20,994—1913
Evaporating surface, sq.ft.	969
Superheating surface, sq.ft.	269
Grate surface (flat) total, sq.ft.	36.2
Economizer, sq.ft.	377
Guaranteed performance:	
Evaporation normally, 5070 lb. water per hour; 3968 lb. steam to be superheated.	
Overall efficiency, per cent.	76
Temperature of superheated steam, deg. F.	625
Gage pressure, lb.	170
Conditions:	
Heating value per pound of coal as fired, minimum B.t.u.	11,500
Permissible residue, per cent.	6
Draft after passing economizer, at least 0.6 to 0.8 in. of water column.	
Feed-water temperature, not less than 95 deg. F.; from economizer, 158 deg. F.	
Coal analysis:	
Average calorific value of coal sampled, 12,145 B.t.u. per pound as fired.	
Results of test:	
Date	Oct. 18, 1913
Load	Normal
Duration of test	4 hr. 2 min.
Temperature of fireroom	57.2 deg. F.
Draft at rear of boiler, in. water column	0.3
Draft after passing economizer, in. water column	0.68
Flue-gas temperature after economizer, deg. F.	405
Flue-gas temperature at rear of boiler, deg. F.	653
Per cent. of CO_2	13.38
Per cent. of O	5.72
Steam pressure, gage, lb.	167
Steam pressure, absolute, lb.	181.7
Temperature of superheated steam, deg. F.	628
Temperature of saturated steam, deg. F.	374
Amount of superheat, deg. F.	253
Total heat of saturated steam (above 32 deg. F.) B.t.u. per lb.	1,197
Heat of superheat, 254 × 0.646 (spec. heat) B.t.u. per lb.	164
Total heat of superheated steam, B.t.u.	1,361
Feed-water temperature to economizer, deg. F.	97.1
Feed-water temperature from economizer, deg. F.	181.3
Heat supplied in boiler, per lb. of superheated steam, B.t.u., 1361 — (97.1 — 32) =	1,295.9
Water evaporated during test, lb.	21,076
Water evaporated per hour, lb.	5,225
Water evaporated per hour per sq.ft. heating surface, lb.	5.39
Coal consumed during test, lb.	2,595
Coal consumed per hour, lb.	643
Coal consumed per hour, per sq.ft. grate surface, lb.	17.8
Water evaporated during test, per pound of coal fired, lb. 5225 ÷ 643	8.14
Heat contained in 1 lb. of saturated steam = 1197 — (97.1 — 32), B.t.u.	1,132.9
Ash and refuse, percentage	9
Heat imparted to saturated steam per lb. of coal as fired, B.t.u., 8.14 × 1131.9 =	9,214
Efficiency of boiler and economizer, referred to saturated steam, per cent., $100 \times \frac{\text{9214 B.t.u. in steam}}{\text{12,145 B.t.u. per lb. coal}}$ =	76

At the time of this test it was impossible to ascertain the steam consumption of the main engine, which would have shown the capacity of the superheater. Therefore, the overall efficiency of steam generation could not then be had. However, a month later the steam consumption of the engine was found to be 2888 lb. of superheated steam per hour. The work done by the superheater may, therefore, be taken at

2888 × 164 B.t.u. = 473,632 B.t.u.

This divided by 643 results in 736.6 additional heat units obtained per pound of coal fired, making the total

9214 + 736.6 = 9950.6 B.t.u.

*Excerpts from an article by Richard Stetefeld, "Eis Und Kalte Industrie."

The heat utilized by the boiler, economizer and superheater, based on coal as fired, is

$$100 \times \frac{9950.6}{12{,}145} = 82 \text{ per cent.}$$

Based on combustible it is

$$100 \times \frac{9950.6}{12{,}145 \times 0.91} = 90 \text{ per cent.}$$

The number of pounds of superheated steam obtained per pound of coal as fired is

$$\frac{9950.6}{1295.9} = 7.68 \text{ per cent.}$$

A repetition of this test was deemed unnecessary because substantially the same results had been secured at a preliminary test.

TEST OF ENGINE COUPLED TO AMMONIA COMPRESSORS

An official trial of the tandem-compound main engine with 17¾ and 29½ by 29½-in. cylinders, of the poppet-valve type, coupled to two 13x23⅝-in. opposed ammonia compressors, was conducted. The reserve engine, a duplicate, was not tested for steam consumption, because the indicator diagrams taken from it under like conditions agreed with the diagrams obtained from the main engine, proving that the power and economy of the two engines in the plant are the same.

Dimensions and Conditions Imposed:	
Diameter of high-pressure cylinder, in.	17.718
Diameter of high-pressure piston rod, in.	3.736
Diameter of high-pressure tail rod, in.	2.749
Diameter of low-pressure cylinder, cold, in.	29.525
Diameter of piston rod, crank end, in.	4.128
Diameter of piston rod, head end, in.	3.736
Stroke, in.	29.527
Revolutions per minute	95
Gage pressure of steam at throttle, lb.	164
Temperature of steam at throttle, deg. F.	572
Cooling surface of surface condenser, sq.ft.	592
Amount of cooling water supplied per hour, cu.ft.	2120
Temperature of cooling water, deg. F.	68
Guarantees:	
Indicated English horsepower of engine, normally	320
Indicated English horsepower of engine, maximum	394
Steam consumption per English i.hp.-hr., lb., at normal load	10.5
Steam consumption per English i.hp.-hr., lb., at maximum load	11.2
Mechanical efficiency of engine, per cent., at normal load	89
at maximum load	90

As for guaranteed efficiency of the exhaust-steam oil separator, the amount of oil remaining per thousand pounds of condensate was not to exceed 0.003 lb.

Results:	
Horsepower indicated	275.9
Revolutions per minute	95
Gage pressure of steam at throttle, lb.	160
Temperature of steam at throttle, deg. F.	576.3
Receiver pressure, inches mercury	5.7
Vacuum, inches mercury	27.6
Duration of test, hr	4.8

The steam consumption of the engine was found to be 2888 lb. per hour. In calculating the indicated horsepower from the diagrams due allowance was made for the expansion of the cylinders under working temperature. The result was 275.9 i.hp. Accordingly, the steam consumption per indicated horsepower-hour during the test was

$$\frac{2888 \text{ lb.}}{275.9} = 10.47 \text{ lb.}$$

The coal consumption per i.hp. per hour was

$$\frac{10.47}{7.68} = 1.363 \text{ lb.}$$

This consumption, it will be noted, was obtained while the pressure of admission and the load were slightly below the figures stipulated in the contract. The normal indicated horsepower of 320 was not developed because the attached ammonia compressors and auxiliary machines required less than 320 i.hp. Diagrams show that the engine is easily capable of developing 320 i.hp., and will not at this load exceed the consumption guaranteed. By lengthening the cutoff the maximum power of 394 i.hp. may be obtained.

TESTS OF COLD STORAGE AND ICE PLANT

Of the two double-acting opposed Borsig ammonia compressors coupled to the engine one and one-half compressor cylinders operate on the cold-storage plant, cooling brine to 14 deg. F., while the fourth compressor cylinder half operates on the ice-making tank, cooling brine to about 19 deg. F. The refrigerating capacity of the one and one-half compressor cylinders was ascertained by measuring with Poncelet nozzles the column of circulating brine cooled per hour through an observed range of temperature. The fourth compressor-cylinder half as well as the single compressor of the reserve engine were indicated to ascertain their working conditions and refrigerating capacity.

Dimensions and Conditions Imposed:	
Diameter of ammonia compressors, in.	12.992
Piston-rod diameter (no tail rod)	3.643
Stroke, in.	23.622
Revolutions per minute	95
Exterior pipe-cooling surface, sq.ft. in brine-cooling tank	3,229
in ice-making tank	1,292
in ammonia liquid cooler	291
in atmospheric ammonia condenser	4,338
Temperature of circulating brine in tank, deg. F.	14
Temperature of brine in ice tank, normal, deg.	19.4
Temperature, initial, of condenser water, deg.	50.0
Temperature of liquefaction of ammonia, deg.	71.6
Temperature of under-cooled ammonia, deg.	52.7
Guaranteed tons refrigerating capacity of one compressor when cooling brine to 14 deg. F. equals $\frac{185{,}000 \text{ calories per hour}}{3023.95} = 61.2$ tons.	
Same for one and one-half compressor cylinders, tons	91.8
Guaranteed daily ice-making capacity of one compressor-cylinder half from distilled water cooled to 53.6 deg. F., lb.	44,000
Indicated horsepower of one compressor when cooling brine to 14 deg. F.	57.2
Indicated horsepower of one compressor-cylinder half, cooling brine to 19.4 deg. F.	33
Consumption of 50-deg. F. condenser water, gal. per min.	265

The power consumption expressed in horsepower was as follows:

One and one-half compressor cylinders, cooling brine	85.8
One-half compressor cylinder making ice	33
Steam-condenser pump	4.4
Water-supply pump	20.5
Brine-circulating pump	9.7
Brine-circulating pump	8.8
Generator, $\frac{70 \text{ kw.} \times 1.34}{0.925} =$	101.5
Losses in transmission	29
Total	292.7

Dividing by 0.89, the mechanical efficiency, the indicated engine horsepower necessary according to the guarantee is 328.4.

Capacity of brine-circulating pump for brine-wetted air-cooler, per min., gal.	396.5
for frosted air-cooler, per min. gal.	396.5
Current consumption of ice crane, kw.	2.5
Current consumption of fans, kw.	13.5

Results of tests made Oct. 18 to 21, and Nov. 21, 1913. Refrigerating capacity of one and one-half compressor cylinders:

Quantity of brine circulated per min., gal.	543
Heat capacity per gal. of salt solution per deg. F. temperature rise, B.t.u. $= \frac{62.35 \text{ lb.} \times 0.946}{7.4805} =$	7.88
Temperature of incoming brine, deg. F.	19.9
Temperature of outgoing brine, deg. F.	13.8
Temperature reduction of brine, deg. F.	6.1
Heat abstracted from brine per min., B.t.u. $543 \times 7.88 \times 6.1 =$	26,100
Corresponding tons of refrigeration performed by one and one-half compressor cylinders, $\frac{26{,}100}{200} =$	130.5

Excess capacity of the one and one-half compressor cylinders over the capacity guaranteed = 130.5 — 91.8 = 38.7 tons, or 42* per cent.

In connection with this brine-cooling test the following interesting temperatures were noted:

Temperature of ammonia at suction pressure, deg. F.	5
Corresponding gage pressure, lb. per sq.in.	19.1
Temperature of ammonia in suction pipe of compressor No. 1, which operates with the same suction pressure in both ends, deg. F.	4.1
Temperature of ammonia in suction pipe of compressor No. 2, only one-half of which operates on the brine tank, deg. F.	6.3
Corresponding gage pressure, lb. per sq.in.	20.2
Temperature of saturated ammonia at discharge pressure, deg. F.	74.6
Corresponding gage pressure, lb. per sq.in.	125.5
Temperature of ammonia in discharge pipe No. 1 compressor, deg. F.	187.7
Extent of superheating, deg. F., 187.7 — 74.6	113.1
Temperature of ammonia in discharge pipe of No. 2 compressor, deg. F.	182.2
Temperature of liquid ammonia leaving aftercooler, deg. F.	54.4
Temperature of ammonia entering brine-cooling coils, deg. F.	14.6
Temperature of ammonia vapor at inlet to atmospheric condensers, deg. F.	148.8
Temperature of liquid ammonia leaving condensers, deg. F.	65.4
Temperature of cooling water, to liquid ammonia cooler, deg. F.	48.9
from liquid ammonia cooler, deg. F.	55.8
leaving ammonia condenser, deg. F.	65.4

Under the above conditions the double compressor (with cylinders Nos. 1 and 2) was running at 94.5 r.p.m., and the

*The amount of this excess capacity suggests the possibility of error in the quantity or specific heat of the brine.

diagrams taken showed the following indicated horsepower:

Compressor No. 1, working full	78.85
Compressor No. 2, one end only	39.16
Total indicated horsepower	118.01

Thus the indicated compressor horsepower per ton of refrigeration in cooling brine to 14 deg. F. was

$$\frac{118.01}{130.5} = 0.904 \text{ per ton,}$$

while the guaranteed power consumption had been equivalent to

$$\frac{57.2}{61.18} = 0.935 \text{ i.hp. per ton refrigeration.}$$

The piston displacement (area × stroke × r.p.m.) of compressor No. 1 is as follows:

$$12.992^2 \times 0.7854 = 132.6 \text{ sq.in.}$$

less one-half the area of the rod or 4.93 equals 132.6 — 4.93 = 127.67 sq.in., which, times the stroke of 23.6 in., is 3013 cu.in.; and this multiplied by 94.5 × 2 strokes = 569,500 cu.in. as the displacement of the cylinder.

The brine-cooling half of compressor No. 2 has a displacement of 273,900 cu.in., and the total for the one and one-half cylinders is 843,400 cu.in. per minute.

Thus the piston displacement per minute per ton of refrigeration when cooling brine to 14 deg. F. was 6419 cu.in.

The piston displacement of the head end of compressor No. 2, making ice, is 3132 cu.in., which times 94.5 strokes gives 296,000 cu.in. per minute, or 40 tons of refrigeration when cooling brine to 14 deg. F.

Since it requires, with the water available, at most 1.6 tons of refrigeration to produce one ton of ice, the ice-making capacity of this compressor-cylinder half is 28.7 tons in twenty-four hours, or 26.3 tons for twenty-two hours' operation daily. The guaranteed capacity of 22 tons is, therefore, exceeded.

The indicated horsepower of the fourth compressor-half when cooling brine to 14 deg. F. was 39.65, equivalent to

$$\frac{39.65}{28.7} = 1.38 \text{ i.hp. per ton of ice,}$$

which corresponds to 1.55 i.hp. in the steam cylinder of the engine.

During the brine-cooling test the cooling water was supplied by the electrically operated volute pump located in the boiler house. The quantity delivered at 1985 r.p.m. was found to be 252 gal. per min. Consequently more than the normal quantity of work was done with less water than the 268 gal. allowed under the guarantee.

The power indicated at the main engine was 283.32 i.hp., and the total power consumption of the establishment was but 9 per cent. more than was guaranteed.

The delivery capacity of the deep-well pump also was checked by noting whether the water level in the receiving basin remained constant while the cooling-water pump was discharging to the condensers at the rate of 268 gal. per min. It was found that even then some water returned to the wells, consequently the well pump was fully up to the capacity guaranteed.

The various features of special interest in the plant are the consistent use of reserve machinery, the obtaining of all power from one economical engine using superheated steam, making ice as a byproduct from exhaust steam, the efficient boiler plant, also the comparatively high rotative speed of the compressors (95 r.p.m., against 80 to 60 in this country). The low coal consumption and low power consumption lead to an unusual economy in the ice-making department. The coal consumption per indicated horsepower-hour was 1.365 lb., or 32.76 lb. per twenty-four hours. This, multiplied by 1.55 i.hp. per ton ice, equals 50.8 lb. Increasing this by 33⅓ per cent. to cover all possible auxiliaries, the total is 67.7 lb. Thus the number of tons of ice made per ton of coal fired is

$$\frac{2000}{67.7} = 29.5.$$

In American plants 10 tons of ice per ton of coal is considered quite satisfactory. It must be remembered in this connection that the initial temperature of the cooling water was 50 deg. F., the suction pressure nearly 20 lb. gage, and the condenser pressure only 126 lb. gage. The remarkable economy of this plant is directly traceable to these favorable operating conditions and to the use of a compound condensing engine using highly superheated steam for driving the compressors and for all the auxiliaries of the plant. The required distilled water for filling the ice cans was obtained only because the refrigerating effect needed to make the ice was only 26 per cent. of the total refrigerating effect produced by the two compressor cylinders. The total amount of steam passing through the engine was 34 tons per day, which, after deducting the water of condensation and other losses, would not have yielded quite enough distilled water for making the maximum of 29.5 tons of ice per day.

An Eight-Cylinder Gasoline Engine*

By Henri G. Chatain†

The object of this paper is to describe and discuss the features of design of an eight-cylinder, four-stroke cycle gasoline engine, which has been developed during a number of years for railway traction.

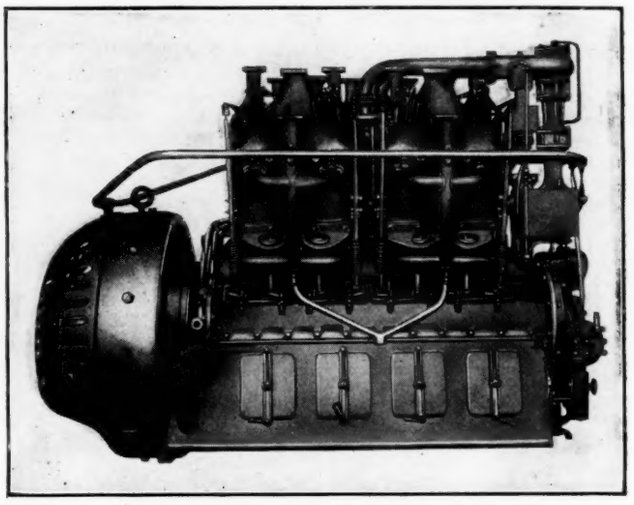


Fig. 1. Side View of Eight-Cylinder Engine

The problem was to design a complete motor car for branch railway service, of sufficient size for seating fifty passengers and a small baggage compartment, etc., and capable of attaining a maximum speed of 50 miles per hour on level track. The first design included an eight-cylinder V-type, 90-deg. engine, with cylinders 7½x8 in., and running normally at 550 r.p.m. This operated commercially in a little car on a Western road until quite recently, when it was destroyed by

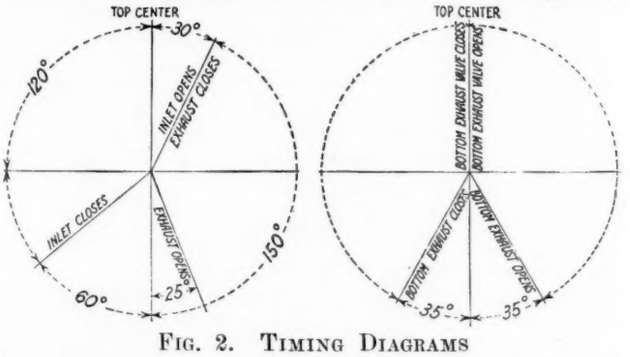


Fig. 2. Timing Diagrams

fire. The second attempt was a larger and heavier engine of essentially the same type. This was run experimentally in service 50,000 miles in one year on various roads, and served its purpose admirably in showing up some glaring defects, among which were the following:

1. The exhaust valves needed regrinding once a week, as they were unduly distorted and burnt.
2. The camshaft and valves were extremely inaccessible.
3. It was uncommercial to make the cylinders sufficiently strong.
4. It was difficult to mount the engine in the car so as to take care of the horizontal component of the reciprocating forces; hence, there was vibration.
5. The total width of the engine was excessive.

With these facts at hand and an ever-increasing demand for more power, the third and present engine, which fulfilled expectations and met conditions, was designed. Fig. 1 gives

*From a paper presented before the Society of Automobile Engineers.

†Engineer, Gas Engine Department, General Electric Co.

a idea of its construction. To overcome the first difficulty mentioned, auxiliary exhaust valves with port entrance to the cylinders were embodied in the design. The cams actuating these valves are so laid out that the valve is entirely free of its seat when the piston passes the port opening. The timing is shown in the diagrams of Fig. 2. This arrangement

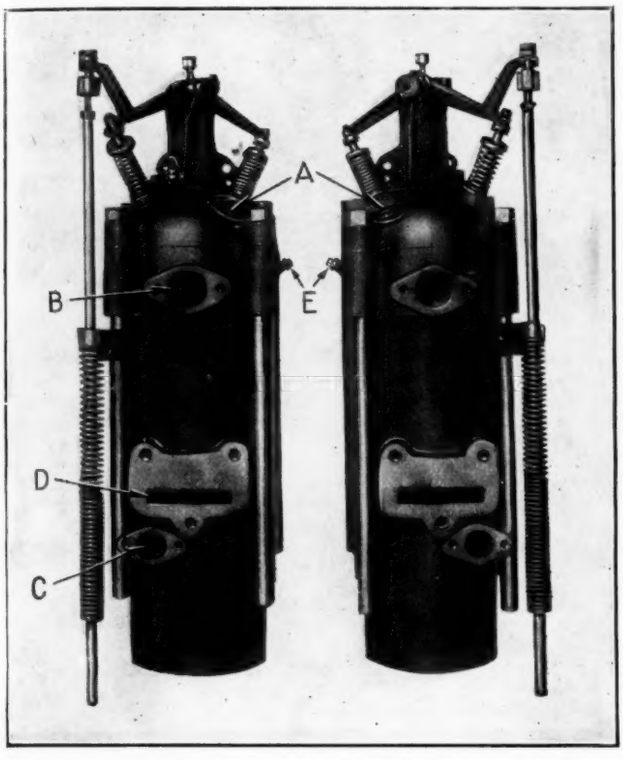


FIG. 3. SHOWING CYLINDER CONSTRUCTION

has worked well, and the auxiliary valves need practically no attention except about once a year. The exhaust valves in the head need to be ground about every 50,000 miles.

The second change was the relocation of the camshaft due to its inaccessibility, and also because the stroke of the engine was to be lengthened. Two externally located camshafts

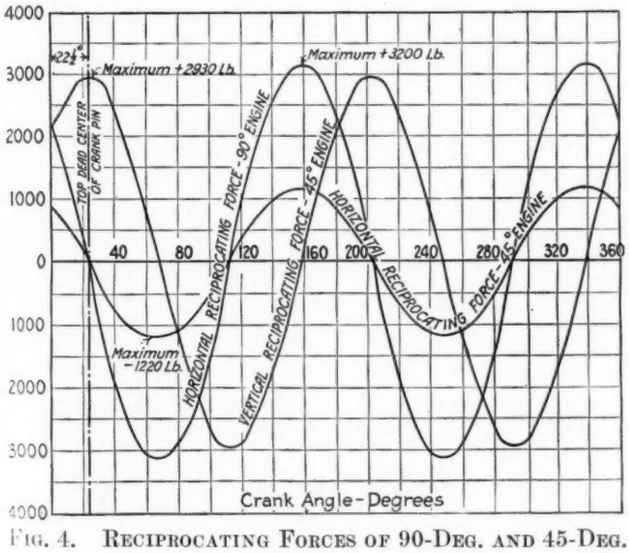


FIG. 4. RECIPROCATING FORCES OF 90-DEG. AND 45-DEG. ENGINES

were decided upon, with the cams, shafts, etc., running in a bath of oil. One actuates the auxiliary exhaust valve, and the other imparts the proper motion to the long push rod extending to the top of the cylinder and connected to the intake and exhaust valves.

In an engine of this size the L-head form of cylinder had little to recommend it. The castings had to be made very heavy, otherwise they would crack; and heavy castings interfere somewhat with the cooling. Extremely strong construction at this point was essential. Fig. 3 shows the form of cylinder construction adopted, namely, a barrel, a head and valves contained therein, held down to the base by long studs. Note that the water-circulating systems of the head and the barrel are distinct. The arrangement of two valves actuated from one cam is quite satisfactory at 500 to 600 r.p.m. With higher speeds the design would be unsatisfactory, as the mass of the moving parts would necessitate unduly high spring pressures. In Fig. 3, A indicates the make-and-break spark plugs, E the air-starter valves, C and B the water inlets, and D the exhaust ports.

A 45-deg. angle between the rows of cylinders was decided upon, to decrease the horizontal component of the reciprocating forces as well as the overall width. Thus each cylinder is set 22½ deg. from the vertical.

The crankshaft has four crankpins, of which the two outer are 180 deg. from the two inner pins and two pistons are attached to each pin. The connecting-rods are mounted side by side on each crank. It was thought advisable to adopt this construction for mechanical simplicity. The piston-pins are held fast in the rods and find their bearings in the piston proper. No bushings are used.

The reciprocating forces for one cylinder along its axis are

$$0.0000284 \times W \times r \times N^2 \left(\cos\theta + \frac{\cos 2\theta}{n}\right) \text{lb.}$$

W = Weight of reciprocating parts = 59 lb.;
r = Crank throw in inches = 5;
N = R.p.m. = 550;
n = Length of connecting-rod divided by throw of crank = 4.5;
θ = Crank angle in the direction of rotation from top dead center of piston.

Fig. 4 shows the reciprocating forces for this engine represented graphically and compared with the 90-deg. cylinders.

The specifications of this engine are as follows:

Cycle		4
Number of cylinders		8
Revolutions per minute		550
Bore		8 in.
Stroke		10 in.
Displacement		503 cu.in.
Valve area:	Inlet	7.07 sq.in.
	Exhaust	4.91 sq.in.
Valve lift		¾ in.
Mean velocity of air in intake pipe		7100 ft. per min.
Exposed radiating surface per cylinder:	Max.	384 sq.in.
	Min.	133 sq.in.
Projected bearing surface:	End	24 sq.in.
	Center	15 sq.in.
Section of connecting-rod:	Max.	2.58 sq.in.
	Min.	1.57 sq.in.
Number of bearings		5
Length of cylinder		23¼ in.
Length of piston		11½ in.
Piston-pin center below piston top		5½ in.
Number of rings		6
Ring size		¾ in.
Ring spacing		¾ in.
Length of connecting-rod, c. to c.		22½ in.
Diameter of piston-pin		2 in.
Diameter of shaft		4 in.
Length of end bearings		6 in.
Length of center bearings		3½ in.
Number of camshaft bearings		4
Length of camshaft bearings		2½ in.
Diameter of camshaft		1½ in.
Cylinder-wall thickness		¾ in.
Water space around cylinder		¾ in.
Height overall		6 ft. 3½ in.
Length less generator		7 ft.
Length including generator		9 ft. 3 in.
Width overall		4 ft. 7½ in.

Alaskan Coal

Last year a sample of coal from the Bering River field in Alaska was tested by the Navy Department and found to be unsatisfactory for the use of the navy. A test has since been made with coal from the Matanuska field, of which Admiral Griffin says:

> Unlike the tests that were made with Bering River coal last year, it was not necessary to hand-pick the Matanuska coal for the purpose of these tests. It was used in the same condition as when delivered, and the results are so satisfactory as to justify the belief that Matanuska coal is in all respects satisfactory for navy use, provided the coal tested is a thorough indication of the general character of the coal in the field.

Secretary Daniels said at a recent hearing that it compared favorably with the best steaming coal that the navy has, and that it was put down at 97 per cent. against 100 for the best coal.

Behavior of Riveted Joints under Stress*

By James E. Howard

The efficiencies of riveted joints under rupturing tensile stresses constitute the values on which working loads are commonly based. Alleged factors of safety are employed, fortunately not less than five on important work, because a fairly good distribution of load is not always characteristic of riveted construction. The most important feature, however, is in the elastic behavior of the joints, which appears to be ignored. As a matter of fact, few riveted structures are free locally from strains which do not exceed the elastic limit of the material. The structures are not necessarily endangered by the presence of such overstrains, as this will depend upon the character of the work to be performed. Multiple-riveted, double butt-strap joints may have a degree of rigidity equal to or even in excess of the solid plate for comparatively low tensile or compressive stresses, but loads ranging from, say, 15,000 to 25,000 lb. per sq.in. commonly show a material divergence in the behavior of a joint over that of the solid plate.

When frictional resistance contributes toward the initial rigidity of the joint, it is uncertain whether the favorable showing of the joint in the laboratory test is realized and maintained under service conditions. Vibratory effects and changes in temperature seem likely to cause a creeping of the plates and disturb the initial state of the different plies, when taken in conjunction with a constant load, or, it may be more marked, in the case of alternate stresses.

Referring specifically to the strength of those parts on which reliance is placed in the design of a riveted joint, first comes the tensile strength of the plate taken as a whole; next to this the strength of the steel between the rivet holes—that is, on the net section. On the latter section the strength per unit of area is not the same with different pitches. It may be greater or less than that accredited to the gross section per square inch. It is also modified according to whether the holes are drilled or punched, and may be greater in one case or the other according to the distance from rivet hole to rivet hole. It is not likely that the strength with punched holes will be greater than with drilled holes in practice, since very closely pitched work is required to bring about such a result. The reason, however, that a punched plate may display greater strength than a drilled one is found in the hardening of the steel by the punch and die at the sides of the holes.

The tensile strength on the net section of the plate is usually greater than on a strip of uniform width several inches in length. The increased area of metal on each side of the center line passing through the rivet holes has a reinforcing effect on the net section of the plate. This gain in strength is a substantial one in single-riveted work, and in multiple riveting when the same pitch is maintained in the different rows. The reinforcement is greater in close-pitched than in wide-pitched riveting, and is at the sides of the holes, but if they are very far apart there results a loss instead of a gain. The reinforcement is therefore not a fixed amount, but depends upon the proportions of the joint.

When the pitch has been considerably increased, as in butt joints with double covers, in which one strap is considerably wider than the other, joints which fail by the rupture of the plate not infrequently show a diminution in strength on the net section, and the plate tears apart at the outside row of rivet holes. The presence of a few rivets, widely spaced, in the outside row promotes tearing of the plate, the line of rupture starting at a rivet hole and reaching an advanced stage before the plate at the middle of the pitch is separated.

Tests on staggered riveting have shown a tendency for the plate to draw down along shearing planes, obliquely to the direction of pull, encountering a rivet in the adjacent row. That is, the design of the joint was such that those in adjacent rows occupied critical positions with reference to each other, and while the zigzag path from one row to the other was longer than from rivet to rivet of the same row, nevertheless the plate showed a preference to fracture along this greater length, and the interposition of rivets in the second row in critical places was a probable source of weakness.

Chain-riveted work creates a favorable impression when observing and comparing the behavior of different types of joints under test. The distance between rows in chain riveting admits of being very much reduced over current practice without impairing the ultimate strength of the joint.

It will be of interest to refer to the strength of riveted joints at higher temperatures. Under exceptional circumstances the joints of steam boilers might be exposed to temperatures considerably above that due to the steam pressure. Joints have been tested up to a temperature of 700 deg. F., and the strength was found to follow the law which governs that of plain steel bars at different temperatures. There was a drop in strength at 200 deg., followed by an increase, which reached a maximum at about 500 deg., after which the strength fell off. Among the several joints tested at 500 deg. the maximum gain over the cold joints was 27.6 per cent. The shearing strength of the rivets showed an increase at the higher temperatures. Furthermore, it was found that joints which were overstrained at these higher temperatures, even beyond the limits of duplicate cold tests, when subsequently tested to destruction at ordinary temperatures, retained substantially the strength which they had when hot. There was some loss in the ductility of the steel, but without approaching a state of brittleness.

So much for the ultimate strength of riveted joints. Attention must be given the behavior of the joints under stress and whether the working loads are constant or variable, direct or reversed stresses, and in the case of repeated stresses, how many repetitions there will be and the maximum stresses involved. The examinations of some stress-strain curves prepared from earlier tests shows that the joints in general take a wide departure from the curve representing the solid plate, this being noticeable at 15,000 lb. per sq.in., and in some joints as early as at 10,000 lb. This was true with joints having efficiencies of 70 to 80 per cent. Among the joints thus compared were double- and triple-riveted butt joints and quintuple joints in which the inner butt strap was wider than the outer one.

Under 15,000 lb. per sq.in. the joints, in general, displayed an extension one and a half times to over twice the extension of the solid plate. These joints were of the types which are used in steam boiler construction. Observations on the behavior of double-riveted lap joints on some steam boilers which had been in service showed greater extension across the longitudinal seams at the middle of the sheets than in the vicinity of the girth seams.

It is of interest whether riveted seams retain their primitive state under prolonged service stresses or whether they do not slip and eventually display increased extensions under lower loads than suggested by the laboratory tests. From the limited number of observations made it cannot be said that the rigidity of joints on actual structures is greater than would be expected. If there is a difference, they are probably less rigid in actual structures.

The frictional resistance due to the shrinkage of the rivets is apparently a factor in the early behavior of a joint. Whether this force drawing the plates together is acting to its full extent will depend upon the manner in which the riveting is done. A limited range in temperature in cooling is sufficient to apply a contractile force equal to the elastic limit of the rivet metal. But since the hot rivet metal has a very low elastic limit it is necessary to hold the plates together firmly until the rivet has cooled to nearly its final temperature. This requirement is an obstacle to rapid driving, but full efficiency in frictional resistance between the plates requires its observance.

*From a paper presented at the twenty-second general meeting of the Society of Naval Architects and Marine Engineers.

BOOKS RECEIVED

CONCRETE PILE STANDARDS. By Huntley Abbott. Published by Huntley Abbott, 11 Pine St., New York City. Paper; 59 pages, 9x12 in.; illustrated. Price 50 cents.

DRAKE'S TELEPHONE HANDBOOK. By D. P. Moreton. Frederick J. Drake, Chicago, Ill. Cloth; 286 pages, 4x6½ in.; 161 illustrations. Price, $1.

A Coal Treatment Alleged to Be Advantageous—There has recently been in Germany quite a flood of preparations put upon the market for the purpose of making a brew in which coal or coke is to be wetted before being put upon the fire. The alleged result of using these preparations is that the coal burns more readily and that there is a great saving in the amount of fuel required. Herr T. Oryng, of the laboratory of the Berlin Fermentation Institute, has analyzed a number of these preparations and found them to consist of various salts, such as sulphate of magnesia, sulphate of soda, common salt, nitrate of soda, and so on, generally with a small proportion of oxide of iron. He concludes that they cannot have the effects attributed to them.—Foreign Exchange.